Lazear and Rogers

Water Rate Tests of a Corliss Engine using Superheated Steam

Mechanical Engineering

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WATER-RATE TESTS OF A CORLISS ENGINE USING SUPERHEATED STEAM

BY

Weston Bross Lazear
Franklin Griffith Rogers

THESIS FOR THE DEGREE OF BACHELOR OF SCIENCE
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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

WESTON BROSS LAZEAR and FRANKLIN GRIFFITH ROGERS

ENTITLED WATER-RATE TESTS OF A CORLISS ENGINE

USING SUPERHEATED STEAM

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE

OF Bachelor of Science in Mechanical Engineering

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WATER RATE OF A CORLISS ENGINE
using
SUPERHEATED STEAM.

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INTRODUCTION.

It is the purpose of this thesis to determine, if possible, the effect of superheated steam upon the water-rate of slow speed engines. Experiments have been performed, the results of which are herein embodied, upon a Corliss engine as a standard type of slow speed engine. A brief summary of the results of other experimenters, upon the same subject, has also been included, as well as the theoretical consideration of the thermodynamic principles involved.

It has been found advisable to divide the subject, for convenience, into three main divisions, as shown in outline on the previous page.

WATER RATE OF A CORLISS ENGINE using SUPERHEATED STEAM.

Part 1. Historical.

Section 1. Early history and performance of engines using superheated steam.

The economy to be derived from the use of superheated steam has long been realized by engineers. As far as the authors could ascertain, the earliest date at which experiments were made with superheated steam was 1828. It was in that year that Richard Trevithick[1] first turned his attention to the subject. His engineer, Cap't. Gregory, wishing to rival the record performance of a steam engine in a neighboring mine, enclosed the cylinders and steam pipes of his own engine, in the Cornwall Mines, in a casing of brickwork, building a fire-grate beneath and flues around them. Surprising results were obtained. The duty, then measured in millions of foot-pounds per bushel of coal (84 lbs.) was raised from 41 to 63, or an increase of over 50%. Trevithick found by running two tests of 24 hours each, and the load the same in each case, that with five bushels of coal burned under the cylinder, sixty-seven (67) bushels were required under the boilers, or a total of seventy-two (72) bushels, while with no coal burned under the cylinder one hundred and eight (108) bushels were required under the boiler. This

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(1) Thurston, in a paper before the American Society of M.E's.

shows a saving of one-third of the total amount of coal burn- by the introduction of superheating.

For thirty years Trevithick had been trying to do away with cylinder condensation, for in 1802, working with Vivian, he built a non-condensing engine with the cylinder inside of the boiler. This was an effort using more the idea of a steam jacket than of superheated steam. However, in 1832 he had patented a vertical tubular boiler, combined with a superheater. The boiler consisted of a series of vertical pipes partly filled with water, arranged in a circle and fitted at the top and bottom into rings or headers. The superheater pipes were placed vertically in the center of the circle, extending down over the fire, the steam being conducted from the upper header of the boiler through the superheater pipes to the engine cylinder. This cylinder was also jacketed using the waste gases from the boiler.

After the experiments of Trevithick little attention seems to have been paid to superheated steam until about 1850, when many plans were proposed for superheaters and steam dryers of various forms, both attached to the boiler flues and separately fired.

Towardthe end of 1853 Hirn took out a patent for "An apparatus designed to superheat steam for engines, or a Hyper- thermo- generator". In this the steam mixture from

boiler was heated up by the tubes through which passed the hot gases from the furnace. With the boiler pressure at 55 lbs. and steam superheated from 410 to 490 degrees Fahr. he obtained economies of from 20% to 47%.[3]

History shows that about 1855-60 superheated steam began to be experimented upon in marine work. Among the first to test the economy was Mr. John Penn of England, who in 1857[4] fitted superheating apparatus in the steam ship "Valletta" of the Peninsula & Oriental Co. The apparatus consisted of two horizontal sections, each of forty-four (44) 2 inch tubes, and was located in the path of escaping gases from the boiler. Mr. Penn gives the following proportions for the engine, boiler and superheater:-

Engines.	260 H.P. (Nominal)
Boiler heating surface.	19 sq. ft. per H.P.
Superheating surface.	2.75 sq. ft. per H.P.
Steam pressure.	20 lbs. gauge.
Degrees of superheat.	100 (approx.)

The immediate result of the introduction of superheated steam was the saving of 20% in the fuel consumption.

About this same time Mr. J.N. Ryder, also of England, experimented with the Messrs. Parson & Pilgrim apparatus as fitted in marine boilers.[5] This apparatus consisted

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(3) William Ripper, "Steam Engine Theory and Practice". page 147.

(4) "Steam and Steam Engines".- Prof. Jamison. page 171.

(5) Proc. Inst. of Mech. Eng's.- England, 1860, Vol. 19.

of two horse-shoe pipes fixed over the grate of the boiler. Steamers "Osprey" and "Swift" showed a saving of 30% to 40% in fuel over that when saturated steam was used. On H.M.S. Tug "Buster", 25% economy was obtained as a mean of thirty-seven trials, using steam at 8.5 lbs. pressure and a superheat of 150 degrees.

Mr. Ryder also experimented with a superheater designed by Mr. David Patridge. This consisted of a cylindrical shell fitted with tubes, and placed vertically in the uptake of a boiler. The gases passed through the tubes and the steam surrounded them. H.M.S. "Dee", fitted with this apparatus, was tested several months by the admiralty and showed and economy of fuel consumption of from 20% to 25%.

Superheaters continued to be used, more or less, especially in marine work, until about 1870, after which they were practically abandoned. Mechanical difficulties were experienced when superheating was carriod beyond 400 to 500 degrees Fahr., in the form of scored cylinders and valve faces, for the lubricant than in use had too low a flash point to withstand the high temperatures. Also it was found that the deposit on the inside of the superheater tubes, through the use of salt water, resulted in the burning out of the tubes. This was before the days of the surface condenser. Steam of higher pressures and higher normal temperatures at this time attracted the attention of engineers, and they gave up , for the time being, the idea of using superheated steam.

Section 2. Economy of modern types of slow-speed engines using superheated steam.

Superheated steam has been growing in favor for about the last ten years. In this country the persistant demands for cheaper power and the competition of gas engines and other forms of motors, has lead to thouse of moderate degrees of superheat. The old trouble experienced with lubrication has been very largely eliminated by producing a cylinder oil having a high flash point, which will stand the high temperatures In Europe, superheated steam is extensively employed, owing to the high price of coal, and a specially designed type of "Superheated steam engine" has resulted, of which the well known Sulzer and Schmidt engines are examples. In these engines all sliding parts in contact with the steam have been reduced by the use of "poppet" or lift valves.

Numerous trials have been made by different experimenters to determine the effect of superheated steam upon various conditions. The results of the few of the more important are set forth in the following discussion.

Mr. Barrus experimented to determine the effect of superheating upon cylinder condensation. He found a reduction in the cylinder condensation of 8.7% for an average of 41 degrees of superheat, with actual saving in the feed water of 8.5%. Table 1. shows the results of his experiments.(6)

Willans, in an effort to determine the gain

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(6) Barrus, "Engines Tests", page 266.

TABLE I.

EFFECT OF SUPERHEATING ON CYLINDER CONDENSATION.

No.	Description.	Degree of Superheat.	Prop. Cyl. Condens'n.	Same for Sat. St'm.	% Reduced.
1.	750 H.P. Corliss. Cond'g.	24	.186	.307	12.7
2.	750 H.P. Corliss. Cond'g.	24	.164	.261	9.7
3.	500 H.P. Corliss. Cond'g.	25	.234	.255	2.1
4.	350 H.P. Corliss. Cond'g.	37	.181	.243	6.2
5.	350 H.P. Corliss. Cond'g.	37	.253	.334	8.1
6.	600 H.P. Corliss. Cond'g.	59	.105	.215	11.1
7.	300 H.P. Corliss Non-con.	82	.053	.167	11.4

in the economy with superheated steam with various initial pressures, made several experiments with his central valve engine. His results[7] are given in Table II. reference to which shows that the greatest gain in economy is to be expected with the lower pressures. They show, however, that the gain per degree of superheat remains practically constant.

A study of the trial made by Bryan Donkin on his engines shows the effect of superheating with different ratios of expansion. His results, given in Table III. seem to show that, increasing the ratio of expansion, i.e. making the cut-off earlier, allows of a greater percentage of saving for the same degree of superheat. The valuable feature of the experiments is, that they were not only made with extreme care, but the same engine engine was arranged to work under the different conditions.

The saving due to different degrees of superheat is further shown by tests made on a simple condensing Corliss engine, 18.5" by 34.4" running at 74 R.P.M. the results of which appear in Table IV.[8] As might be supposed, the gain in the economy of steam consumption is greater for higher degrees of superheat, but the curve shown on page 13a, plotted from the results given in the table, show that the gain per degree of superheat remains almost constant.

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(7) Ewing, "The Steam Engine", page 166.

(8) Zeitschrift des Verein, vol. 43, p. 1519.

TABLE II.

WILLANS CONDENSING ENGINE.

WITH AND WITHOUT SUPERHEATING

AT VARIOUS PRESSURES.

Initial Pressure.	45 lbs.		35 lbs.		25 lbs.	
Superheated?	No.	Yes.	No.	Yes.	No.	Yes.
Initial Temper.	274	305	255	299	239	289
Degree of Sup't.		31		40		50
Water Rate.	26.7	24.6	28.9	26.4	30.0	26.4
Per-cent gain.		7.8		8.6		12.0
Gain per Deg. S't.		.25		.22		.24

TABLE III.

DONKIN ENGINE.

SUPERHEATING AT DIFFERENT RATIOS OF EXPANSION.
BOILER PRESSURE-50 LBS. R.P.M. 220.

No.	Ratio of Expansion.	Temp. of Steam.	Deg. of Super't.	Water Rate.	% Gain.	% Gain Deg.S't.	I.H.P.
1	1/8	297		55.90			1.98
2		391	54	38.50	31.1	.577	2.50
3	3/16	299		50.54			2.56
4		349	50	41.45	18.5	.370	2.75
5	5/16	297		49.14			3.00
6		347	51	41.00	16.6	.325	3.20
7	1/2	299		46.03			4.14
8		349	50	41.33	10.2	.204	4.22
9	3/4	297		47.67			4.80
10		346	50	44.21	7.24	.145	4.58
11	1/4	298		80.72			1.39
12		348	50	64.00	20.7	.414	1.43
13	5/16	299		74.55			1.75
14		349	50	56.18	24.6	.492	1.91
15	1/2	300		62.03			2.71
16		350	50	50.92	17.9	.359	3.00
17	3/4	298		60.39			3.34
18		347	48	53.39	11.5	.225	3.40
19	1/5	296		46.69			7.17
20		334	38	41.76	10.5	.278	7.02
21	1/12	296		43.58			4.22
22		348	52	26.93	38.2	.735	4.41
23	1/8	295		42.90			4.73
24		363	68	28.45	33.2	.495	5.17
25	3/16	302		40.65			5.64
26		347	45	31.24	23.2	.516	6.30
27	1/4	294		38.64			6.37
28		316	52	31.20	19.2	.370	6.95
29	5/16	297		35.56			7.58
30		313	16	32.87	10.8	.675	7.46
31	3/8	297		36.39			7.73
32		336	39	31.95	12.2	.313	8.59
33	1/2	294		38.30			9.14
34		326	31	33.67	12.1	.390	8.93

1 to 10 inc. Single Acting - Condensing.

10 to 18 inc. Single Acting - Non-condensing.

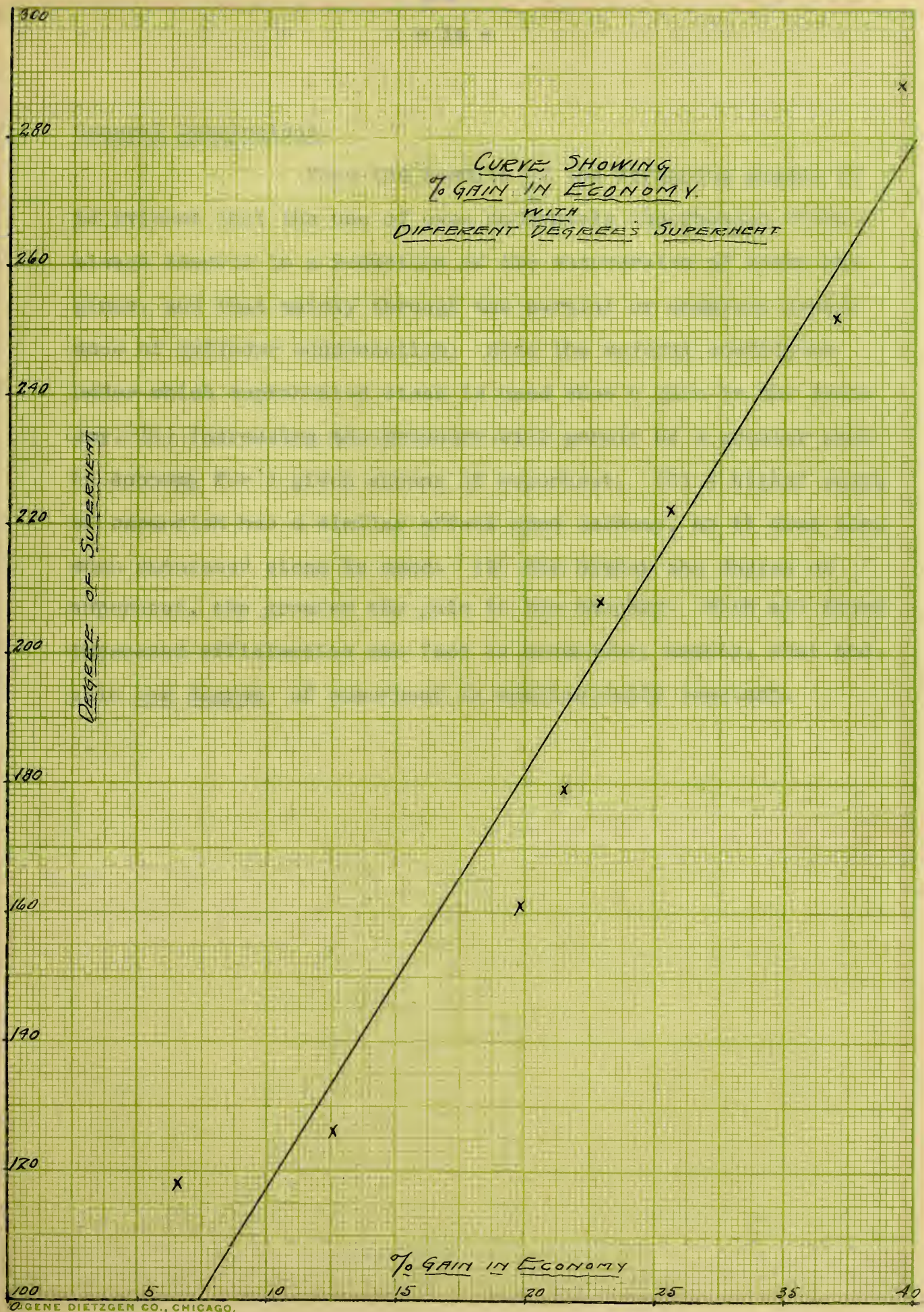
19 to 20 inc. Double Acting - Non-condensing.

21 to 34 inc. Double Acting - Condensing.

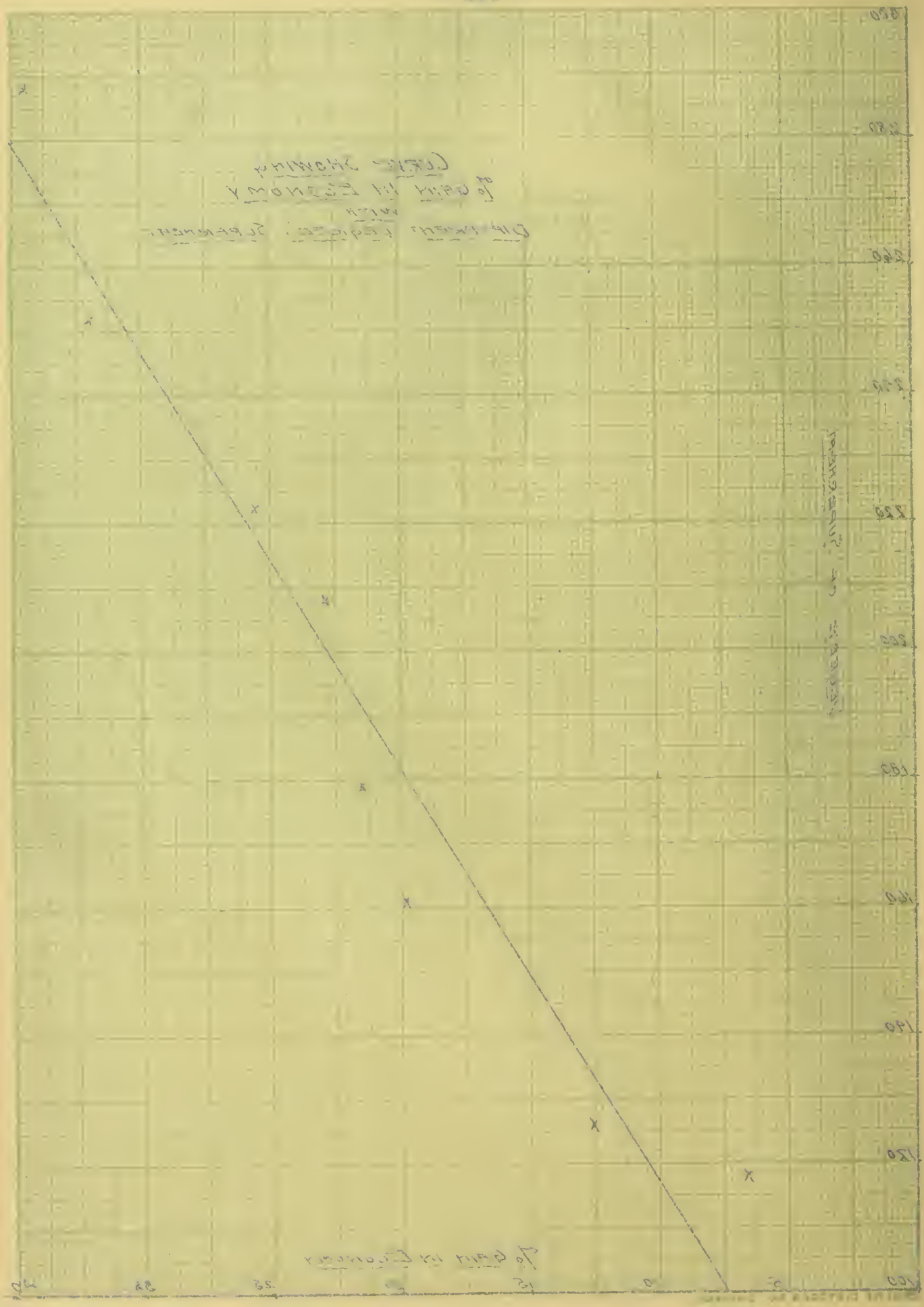
TABLE IV.

EFFECT OF DIFFERENT DEGREE OF SUPERHEAT.

No.	I.H.P.	Temp. of Steam.	Deg. of Super't.	Boiler Press.	Water Rate.	% Gain.	% gain Deg. S't.
1.	29.79	277	78.0	21.0	30.1		
2.	29.03	316	118.5	20.6	28.1	6.65	.056
3.	37.58	336	126.5	25.4	26.3	12.65	.100
4.	44.88	380	161.5	28.3	24.1	19.90	.123
5.	41.87	392	179.5	26.9	23.6	21.60	.120
6.	34.30	412	208.8	22.9	23.2	22.90	.110
7.	37.65	432	222.0	25.4	22.4	25.60	.115
8.	38.90	447	236.0	26.1	22.4	25.60	.108
9.	46.45	473	252.0	28.3	20.5	32.00	.124
10.	49.57	509.5	288.0	29.9	19.7	34.60	.120



EUGENE DIETZGEN CO., CHICAGO.



General Conclusions.

From the results of the foregoing pages, it is evident that the use of even moderately superheated steam always results in a reduction of the water-rates of steam engines, and that mainly through the partial or complete avoidance of cylinder condensation. Also the several conditions under which supreheated steam is used show a gain in the economy. (1) Increasing the pressure will permit of a greater gain in economy for a given amount of superheat. (2) A higher ratio of expansion has a similar effect upon economy, as it does also when saturated steam is used. (3) The higher the degree of superheat, the greater the gain in the economy. With all these increased efficiencies one fact is prominent, namely, that the gain per degree of superheat is substantially constant.

Part II. Theoretical.

Section 1,(a). Clausius cycle for superheated steam.

The Institute of Civil Engineers has established as the ideal standard of comparison for steam engines the "Clausius" cycle, sometimes known as the "Rankine" cycle. This differs from the ideal Carnot cycle only in the substitution for the ideal adiabatic compression, a process which in the actual steam plant occurs in the boiler, vis., heating from the exhaust temperature to the boiling point.

The representation of the ideal Clausius on the pv-plane is given in fig.1, page 16. Four assumptions are made necessary:-

(1) That steam is admitted at constant pressure, so that a-b is a straight line parallel to OV.

(2) That the expansion line b-c is an adiabatic curve, and that the expansion is "complete".(10)

(3) That steam is exhausted against a constant back pressure, so that c-d is a straight line parallel to OV.

(4) That the ordinary compression curve be replaced by a vertical straight line d-a.

The processes on the TØ-plane for the Clausius cycle using superheated steam are as follows:- (See fig.2, page 16.)

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(10) By "complete" expansion is meant the expansion down to a pressure equal to the back pressure.

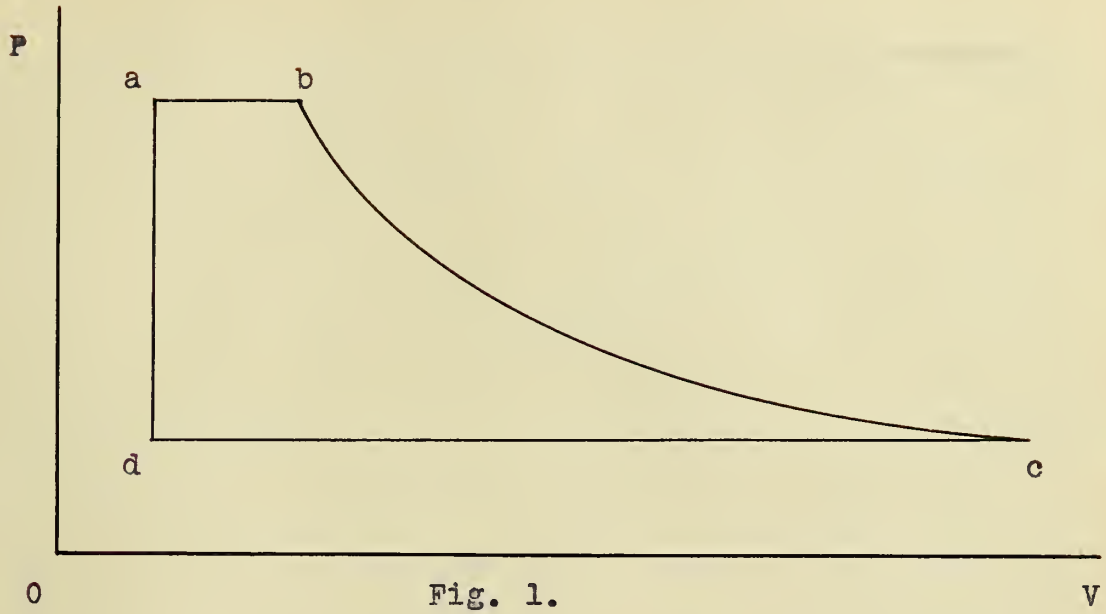

Fig. 1.

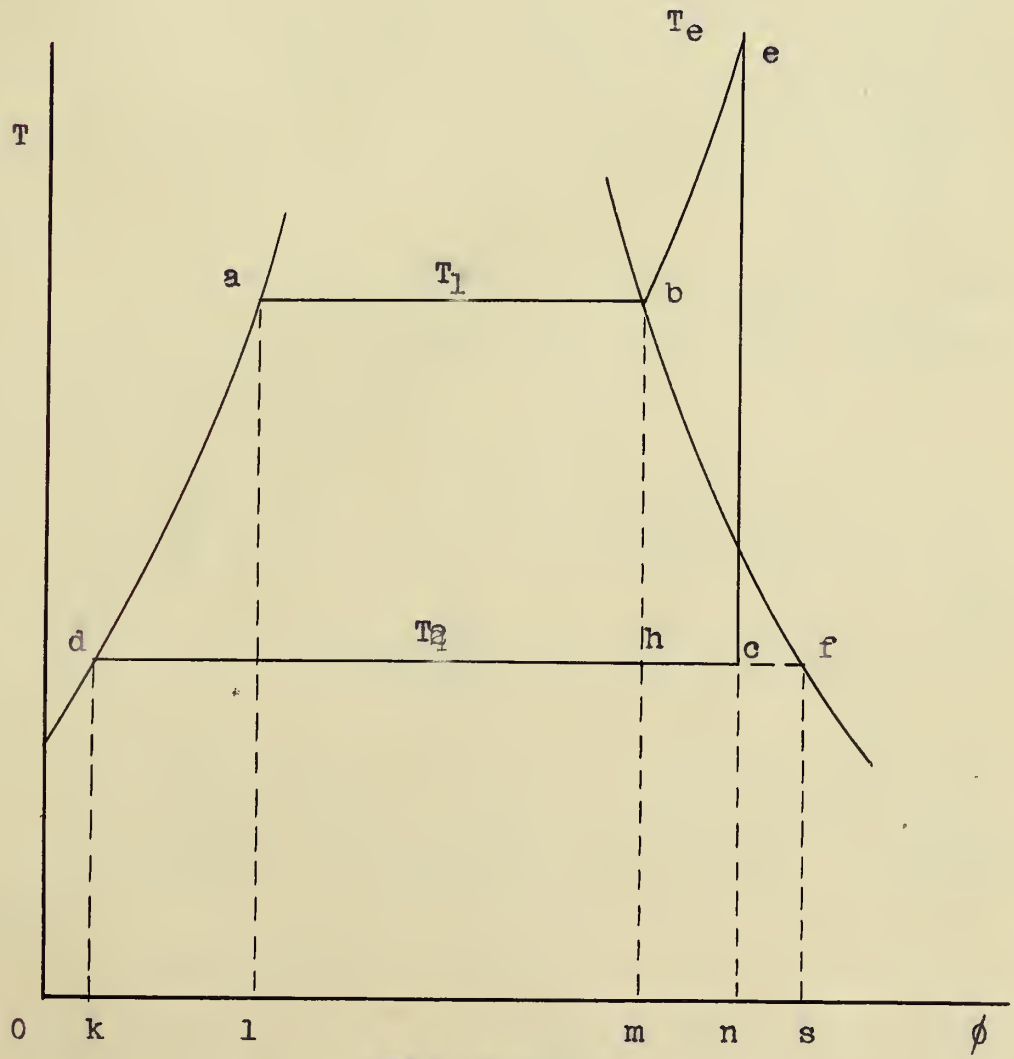

Fig. 2.

From d to a the temperature of the water is raised from the exhaust temperature T_1, to the boiling point T_2, then the isothermal evaporation follows from a to b, or evaporation at constant temperature. At b the water has all been converted into steam, and from b to e occurs the superheating at constant pressure, with increase of temperature from T_2 to T_e. The steam then expands adiabatically from e to c, the temperature at the same time dropping from T_e to T_1, and if the point c is below the saturation curve, some of the steam has been condensed during the expansion. From the state c the medium is compressed and heat is thrown into the condenser or hot-well, until in the state d the medium is completely liquefied. The medium is now in the same condition in which it started.

Let Q represent the heat involved in any change of state, and the subscripts ab, bc, etc., designate the particular change of state. Also

let M = weight of the medium used.

q_1= heat of the liquid at the abs. temperature T_1.

q_2= " " " " " " " " T_2.

r_1= " " vaporization at temperature T_1.

r_2= " " " " " T_2.

C_p= specific heat of superheated steam at constant (pressure.

x_c= quality of the steam at c = KN/KS.

then

$$Q_{da} = M(q_1 - q_2)$$

$$Q_{ab} = Mr_1$$

$$Q_{be} = MC_p(T_e - T_2)$$

$$Q_{ec} = 0$$

$$Q_{cd} = Mr_1x_c$$

and the work of the cycle is = J (total heat of the cycle),

$$\text{or} \quad W = JM\left[q_1 - q_2 + r_1 + C_p(T_e - T_2) - r_2x_c\right]$$

If the efficiency of the cycle is the ratio of the total work of the cycle to the work put in in the form of heat, then the efficiency is

$$= 1 - \frac{r_2x_c}{q_1 - q_2 + r_1 + C_p(T_e - T_1)}$$

The work and efficiency of the cycle, using saturated instead of superheated steam, would be similar, but the term

$$C_p(T_e - T_1)$$

would be omitted, and "x_c" would have a value (= dh/df) smaller than (dc/df) when superheated steam was used, as will be seen by referring to fig.2.

The extra area, hbec, added to the cycle by the superheat, requires the extra heat represented by the area mben, and thus the efficiency of the superheat is the ratio of the area hbec to the area mben. It is very evident from fig.2 that the gain in the theoretical efficiency of the complete cycle due to superheating is really slight, and that the real economy to be derived is not from a theoretical consideration of the heat involved, but from the actual effect upon

cylinder condensation, as found by experiment.

To illustrate the above discussion with a numerical problem:- Suppose water at 120° Fahr. (T_2= 580° Fahr.abs.) is heated in a boiler to a pressure of 115 lbs., the temperature corresponding being 800° F.abs.(T_1). It is then evaporated and superheated to 1100° F.abs.(T_e), or 1100 - 800 = 300° of superheat. It then expands adiabatically to the original temperature of 580° F.abs., and is exhausted into the hot-well. From the steam tables

$$q_1 = 310.9 \qquad r_1 = 874.7$$

$$q_2 = 88.1 \qquad r_2 = 1030.4$$

also $x_c = 0.90$ and $C_p = .48$ (11)

then the efficiency is

$$= 1 - \frac{1030.4 \times 0.90}{310.9 - 88.1 + 874.7 + .48(1100 - 800)}$$

$$= 25.5\%$$

using superheated steam.

The same cycle, using saturated steam, and the expansion taking place from b instead of e has an efficiency

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(11) For many years the specific heat of superheated steam at constant pressure has been taken as a constant value for all degrees of superheat, but recently experiments have been made to show that this assumption is incorrect. However, no definite equation for a variation in value has been decided upon, and C_p = .48 has generally been accepted.

$$= 1 - \frac{r_2 x_c'}{q_1 - q_2 + r_1} \qquad \text{where } x_c' = .80$$

$$= 1 - \frac{1030.4 \times .80}{310.9 - 88.1 + 874.7} = 24.9\%$$

Thus it shows a gain in the theoretical efficiency of only 0.6% by a superheating in this case of 300°, although the efficiency of the superhaat is about 39%.

It is evident that to have dry steam at the end of expansion, and thus have no condensation, the superheating must be carried to a temperature T_e, such that the point c, in fig.2, will fall on the saturation curve. Further superheating would then result in a loss, for the steam would be superheated at release, which would mean waste heat going to the condenser.

Section 1,(b). Clausius cycle for superheated steam with incomplete expansion.

The pv-diagram of fig.1 shows the expansion carried down to the back pressure line cd. In practice it is not usual to carry the expansion so far, because the extra work obtained does not compensate for the increased cylinder volume required. The ideal pv-diagram is made to appear as in fig.3. In the actual engine the process represented by ch is an irreversible rush of steam from the cylinder to the condenser. This, however, is replaced by an ideal reversible process, cooling at constant volume, which gives the same line on the pv-plane, and therefore leaves the work unchanged. The corresponding Tφ-diagram, using superheated steam with incomplete expansion, and shown in fig.4, differs from the original cycle in having the constant volume curve ch.

In order to have dry saturated steam at release which is so desirable, the end of the adiabatic expansion line ec must lie on the saturation curve. If it lies to the right of that curve, heat will be wasted in the form of superheat thrown into the condenser; if it falls to the left, condensation will result. By comparing the cycle of incomplete expansion with that having expansion complete, shown dotted, it is seen that with a given initial pressure superheating does not have to be carried to as high a degree in a cycle with incomplete expansion, in order to obtain dry satur-

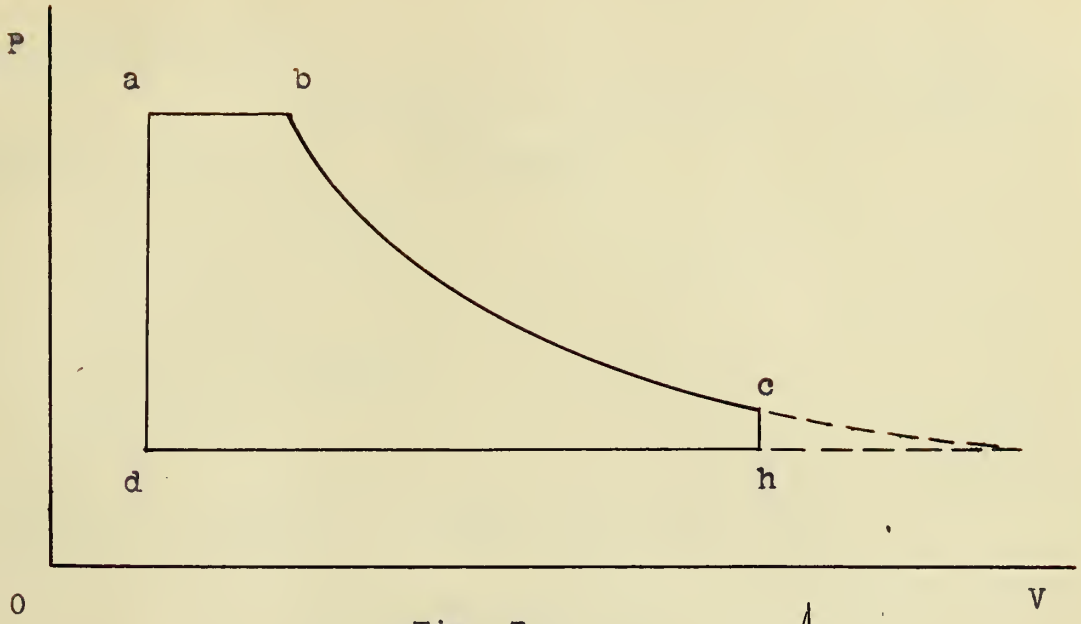

Fig. 3.

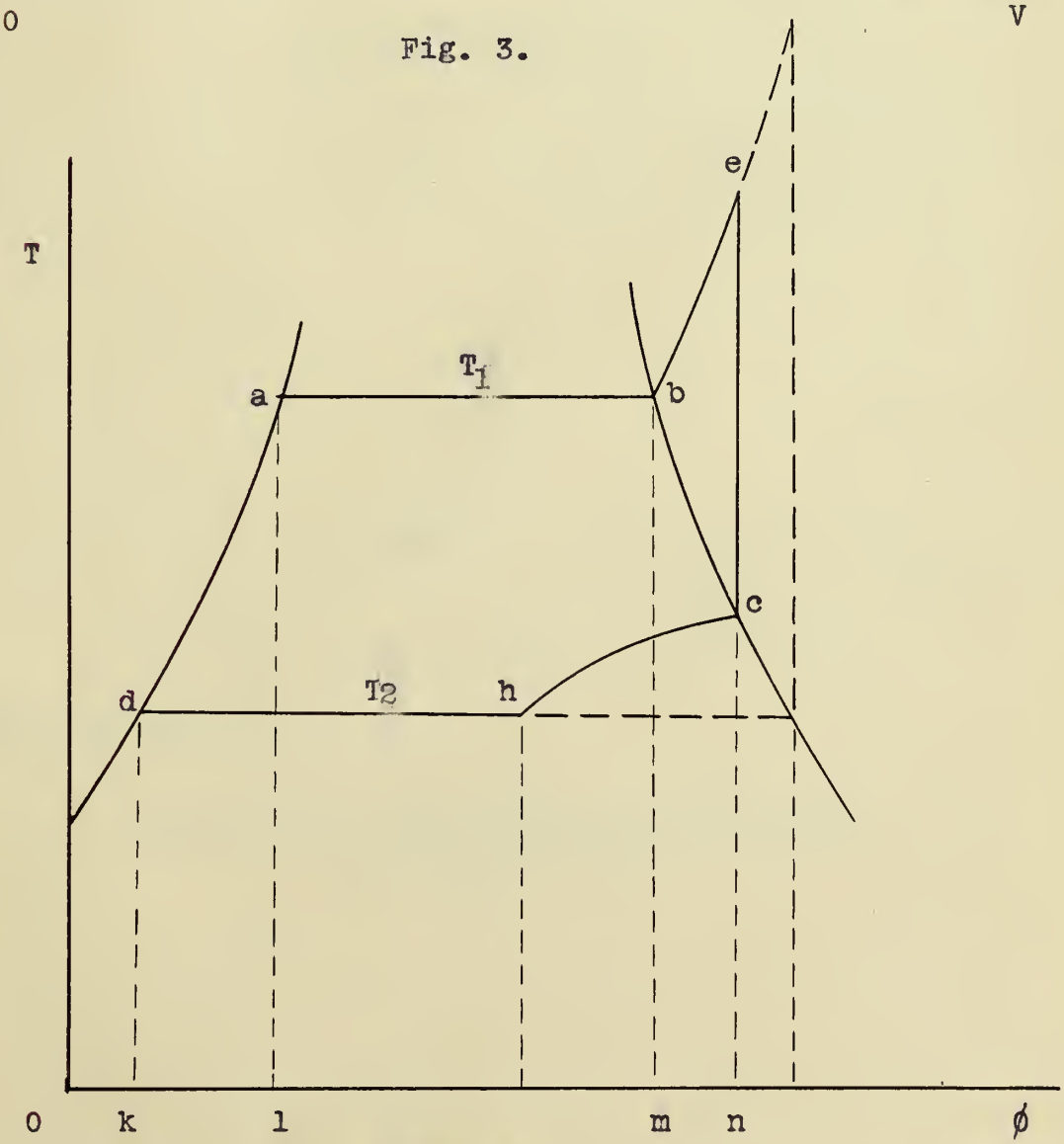

Fig. 4.

ated steam at release.

The work and efficiency of the cycle is derived as follows:- As before

$$Q_{da} = M\,(q_1 - q_2)$$

$$Q_{ab} = Mr_1$$

$$Q_{be} = MC_p\,(T_e - T_2)$$

$$Q_{ec} = 0$$

Now $Q_{ch} = U_c - U_h$ (since the work = 0)

$$= M\,(q_c + x_c\rho_c) - (q_2 + x_h\rho_2)$$

where ρ is the heat equivalent of internal work at the points designated. If c is on the saturation curve, $x_c = 1$.

Also $Q_{hd} = Mx_h r_2$

Then the total work is

$$W = JQ$$

$$= JM\left[q_1 - q_2 + r_1 + C_p(T_e - T_2) - (q_c + x_c\rho_c) + (q_2 + x_h\rho_2) - x_h r_2\right]$$

$$= JM\left[q_1 - q_c + r_1 x_c\rho_c - x_h(r_2 - \rho_2) + C_p(T_e - T_2)\right]$$

and the efficiency is

$$= \frac{q_1 - q_c + r_1 - x_c\rho_c - x_h(r_2 - \rho_2) + C_p(T_e - T_2)}{q_1 - q_2 + r_1 + C_p(T_e - T_2)}$$

Section 2. Ideal steam consumptions.

The work of the Clausius cycle, as found in section 1, for one pound of dry saturated steam was

$$W' = J\left[q_1-q_2+r_1-r_2x_c'\right]$$

Let M denote the weight of steam used per horsepower-hour. Then

$$1 \text{ horsepower-hour} = JM\left[q_1-q_2+r_1-r_2x_c'\right]$$

$$\text{or} \quad 1{,}980{,}000 \text{ ft.lbs.} = JM\left[q_1-q_2+r_1-r_2x_c'\right]$$

$$\text{whence} \quad M = \frac{1{,}980{,}000}{778}\,\frac{1}{q_1-q_2+r_1-r_2x_c'}$$

$$= \frac{2545}{q_1+r_1-q_2-r_2x_c'}$$

The ideal consumption given by this equation is that for complete expansion, and has been proposed as a basis of consumption.(12) Thus for the cycle using superheated steam

$$\underline{M} = \frac{2545}{q_1+r_1-q_2-r_2x_c+C_p(T_e-T_2)}$$

Denoting by $\underline{M}'$ the water consumption of an actual engine, the quotient

$$\frac{M}{M'} = \frac{2545}{M'\left[q_1+r_1-q_2-r_2x_c+C_p(T_e-T_2)\right]}$$

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(12) Trans. A.S.M.E. vol.25, p.920.

gives the efficiency of the actual engine compared with the ideal Clausius engine.

In the problem quoted in section 1 for illustration, the ideal water-rate will be

$$M = \frac{2545}{310.9 + 874.7 - 88.1 - .90\ 1030.4 + .48\ (1100-800)}$$

$$= 8.05 \text{ lbs. per horsepower per hour,}$$

using superheated steam, while using saturated steam

$$M = \frac{2545}{310.9 + 874.7 - 88.1 - .80\ 1030.4} = 9.32 \text{ lbs.}$$

used per housepower per hour. This shows a gain of 1.27 lbs. in the theoretical water-rate by using superheated steam.

Part III. Experimental.

Section 1. Description.

The following tests, to determine the water-rate of the Corliss engine as a type of slow speed engine, were conducted in the Mechanical Engineering Laboratory of the University of Illinois.

Steam. Two mains, one for the saturated steam and one for the superheated steam, extended parallel the full length of the building. The saturated steam was supplied by either the central Power House or the Heinie Boiler located in the laboratory. To obtain superheated steam the saturated steam was conducted from the saturated steam main to a Foster independantly-fired superheater, located at the side of the Heinie boiler, which had a capacity of 400 degrees of superheat. The superheated steam was piped from there back to the superheated steam main, from which it could be taken at the points desired. The exhaust was conducted through a four inch pipe to a 60 H.P. " Wheeler " condenser, located 30 feet from the engine. Atmospheric pressure was maintained in the condenser, and the condensed steam was weighed in a tank mounted on platform scales.

Engine. The engine was a 12" x 24" single cylinder, Corliss engine, made by the Allis Chalmers Co. of Milwaukee, Wis. It was rated at 100 H.P. with a boiler pressure of 120 lbs. and a variable speed of from 60 to 100 R.P.M.

The flywheel was 9'-0" in diameter with a 12" face, and it was used as a pulley to transmit by belt the power developed.

Power. A counter-shaft, located 16'-2" from the main shaft of the engine, carried a pulley 3'-10$\frac{3}{4}$" in diameter, 12" face, which was belted to the flywheel of the engine. The power developed by the engine was measured with an ordinary Prony brake, water cooled, which had a brake pulley 2'-0" in diameter, and a lever arm of 5'-0". The load on the brake was measured directly by platform scales.

Section 2. Methods employed.

Three sets of tests were performed, first set using 0 degrees of superheat, or in other words saturated steam, the second set using steam superheated from 50 to 60 degrees Fahr., and the third using steam of about 100 degrees superheat. Each set of tests consisted of three different loads, 1st no load, 2nd 1/4 load, and 3rd 1/2 load. Readings were taken, simultaneously, every five minutes, the length of te test being one hour for no load and two hours when running under 1/4 and 1/2 loads, using superheated steam.

The degree of superheat at the steam chest was determined by the use of a pressure gauge and thermometer inserted in the top of the chest. The difference between the temperature as shown by the thermometer and the temperature corresponding to the pressure found by the gauge gave the degree of superheat at that point. The steam consumption was determined by condensing the exhaust steam and weighing it.

The power developed in the cylinder (I.H.P.) was measured by means of Crosby indicators, the springs of which were calibrated before being used. The power given out by the engine (B.H.P.) was measured by the Prony brake heretofore referred to, the revolutions of the engine and the counter-shaft being recorded by the revolution counters.

In the preliminary tests it was found that the belt would not transmit more than 50 H.P. although a belt dressing was employed to increase the adhesion between belt

and pulley. It was also found that with the loads used the engine did not take steam fast enough to overcome a large drop in temperature in the steam chest due to radiation. In order to increase the flow of steam to the engine, and thus in crease the temperature of the steam at the chest , the main was tapped at the throttle, and a one inch by-pass was inserted, which was kept open constantly and exhausted outside of the building. Even with this arrangement it was found impossible to obtain mo more than 100 degrees of superheat at the engine, although at the superheater 250 degrees of superheat were maintained.

Section 3.

Results of experiments.

NON-CONDENSING.
STEAM- Saturated
LOAD - 0

No.	P	Wgt. on Scales	B.H.P.	Mech. Eff.	Water-rate B.H.P	Water-rate I.H.P
1						81.4
2	6					54.
3	7					97.
4	9					64.
5	2					52.2
6	6					54.
7	9					
8						79.
9	3					80.3
10	3					90.3
11	3					81.3
1	6					
Aver						73.4
	3					

WATER RATE
OF
CORLISS ENGINE.-

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

NO. OF TEST.- 1

DATE.- April 25, 1907

PRESENTED BY { Weston B. Lagcar. / Franklin G. Rogers

NON-CONDENSING.

STEAM- Saturated

LOAD - 0

No.	Time A.M.	R.P.M.		Superheater			Steam Chest		Degrees of Superh't.	Temperatures.		Cooling Water		Wg't. Steam Condensed	I.H.P	Wg't. on Scales	B.H.P.	Mech. Eff.	Water-rate	
		Engine	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng. Room	Outside	Inflow	Outflow						B.H.P	I.H.P
1	9.00	78	180				137#			68	42	54	59		5.36					81.4
2	9.05	78	180				137					54	59		8.07					54.
3	9.10	79	182				138					58	64		4.49					97.
4	9.15	81	185				138			68	42	58	63		6.82					64.
5	9.20	81	185				132					58	63		8.36					52.2
6	9.25	82	187				132					53	60		8.09					54.
7	9.30	84	192				117			66	42	53	60							
8	9.35	81	183				105					53	58		5.53					79.
9	9.40	78	173				102					53	58		5.43					80.3
10	9.45	83	186				119			65	43	53	58		4.83					90.3
11	9.50	85	192				129					53	62	363#	5.36					81.3
1																				
Average		81	184				126			67	42	55	60	436#/Hr.	6.23					73.4

NON-CONDENSING.

STEAM- Saturated.

LOAD - $\frac{1}{4}$

N.H.P.	Wgt. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
				B.H.P.	I.H.P.
11.4	145#	25.9	82.5	32.8	31.2
20.6		25.8	84.3	38.0	32.0
39.7		25.4	85.0	38.6	33.0
49.5		25.7	87.0	38.2	33.2
59.7		26.4	89.0	37.1	33.0
69.3		26.4	81.8	37.1	30.4
78.4		24.2	85.0	40.5	34.5
89.7		25.0	84.2	39.2	33.0
99.3		25.4	86.8	38.6	33.4
108.9		24.6	85.1	39.0	33.9
11		25.1			
0.0	145	25.4	85.1	37.9	32.8

WATER RATE
OF
CORLISS ENGINE.-

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

NO. OF TEST.- 2.
DATE.- April 25, 1907.

PRESENTED BY { Weston B. Lagear. Franklin G. Rogers

NON-CONDENSING.
STEAM- Saturated.
LOAD - 1/4

No.	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wg't. Steam Condensed	I.H.P	Wg't. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
		Engine	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng. Room	Outside	Inflow	Outflow						B.H.P.	I.H.P.
1	10.00	83	188				142			62	43	52	67		31.4	145#	25.9	82.5	32.8	31.2
2	10.05	83	187				135					52	67		30.6		25.8	84.3	38.0	32.0
3	10.10	82	184				135					53	68		29.7		25.4	85.0	38.6	33.0
4	10.15	82	186				138			65	42	53	68		29.5		25.7	87.0	38.2	33.2
5	10.20	83	191				134					53	68		29.7		26.4	89.0	37.1	33.0
6	10.25	84	191				120					52	68		32.3		26.4	81.8	37.1	30.4
7	10.30	76	175				120			64	42	52	68		28.4		24.2	85.0	40.5	34.5
8	10.35	79	181				132					53	68		29.7		25.0	84.2	39.2	33.0
9	10.40	80	184				141					53	68		29.3		25.4	86.8	38.6	33.4
10	10.45	77	178				132			65	41	52	68		28.9		24.6	85.1	39.0	33.9
11	10.50	80	183				125					56	67	817#			25.1			
Average		81	184				132			64	42	53	68	980#/Hr.	30.0	145	25.4	85.1	37.9	32.8

NON-CONDENSING.

STEAM - *Saturated.*

LOAD - $\frac{1}{2}$.

Wgt. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
			B.H.P.	I.H.P.
290#	47.0	91.0	33.0	29.9
	47.0	89.0	33.0	29.4
	48.8	91.0	31.8	28.9
	53.5	90.5	29.0	26.3
	53.3	84.4	29.1	24.6
	48.8	89.1	31.8	28.4
	49.1	85.6	31.6	27.0
	49.1	89.8	31.6	28.4
	49.1	88.7	31.6	28.0
	51.6	90.8	30.0	27.3
	51.9	88.0	29.9	28.4
290#	49.9	89.0	31.1	27.9

WATER RATE
OF
CORLISS ENGINE.-

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING

NO. OF TEST.- 3.
DATE.- April 25, 1907

PRESENTED BY { Weston B. Lagcar. Franklin G. Rogers

NON-CONDENSING.
STEAM- Saturated.
LOAD - ½.

No.	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wgt. Steam Condensed	I.H.P	Wgt. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
		Engine	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng Room	Outside	Inflow	Outflow						B.H.P	I.H.P
1	11.00	75	170				120			61	41	53	74		51.8	290#	47.0	91.0	33.0	29.9
2	11.05	74	170				129					53	74		52.8		47.0	89.0	33.0	29.4
3	11.10	77	177				129					53	74		53.6		48.8	91.0	31.8	28.9
4	11.15	85	194				130			65	41	53	74		59.1		53.5	90.5	29.0	26.3
5	11.20	84	193				129					53	74		63.2		53.3	84.4	29.1	24.6
6	11.25	78	177				135					53	74		54.7		48.8	89.1	31.8	28.4
7	11.30	78	178				129			62	41	53	74	763#	57.4		49.1	85.6	31.6	27.0
8	11.35	78	178				124					53	74		54.7		49.1	89.8	31.6	28.4
9	11.40	79	178				124					53	74		55.3		49.1	88.7	31.6	28.0
10	11.45	82	187				132			63	41	53	74		56.9		51.6	90.8	30.0	27.3
11	11.50	82	188				135					53	74	530#	54.6		51.9	88.0	29.9	28.4
Average		79	181				129			63	41	53	74	1550#/Hr.	55.7	290#	49.9	89.0	31.1	27.9

NON-CONDENSING.
STEAM - 50° Superheat.
LOAD - 0

No	Wg't. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
				B.H.P.	I.H.P.
1	0				60.0
2					
3					42.5
4					46.4
5					
6					35.4
7					53.6
8					45.5
9					38.2
10					44.2
11					45.3
12					43.7
13					57.7
	0				46.6

WATER RATE
OF
CORLISS ENGINE.-

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

PRESENTED BY { Weston B. Lagcar. Franklin G. Rogers

NO. OF TEST.- 4.
DATE.- May 1, 1907.

NON-CONDENSING.
STEAM- 50° Superheat.
LOAD - 0

No.	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wg't. Steam Condensed	I.H.P.	Wg't. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
		Engine	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng. Room	Outside	Inflow	Outflow						B.H.P.	I.H.P.
1	1.00	81	186	130	123	650	141	409	56	68	51	55	59		5.69	0				60.0
2	1.05	81	186	129	123	630	140	412	59			56	60							
3	1.10	81	186	129	123	640	140	415	62			55	59		8.01					42.5
4	1.15	81	187	125	123	626	134	415	66	67	53	56	60		7.34					46.4
5	1.20	81	187	135	134	610	145	416	60			56	60							
6	1.25	81	187	134	134	616	145	419	63			56	60		9.60					35.4
7	1.30	81	187	130	130	600	142	417	63	67	54	56	61		6.33					53.6
8	1.35	81	187	128	127	570	137	414	63			56	60		7.47					45.5
9	1.40	81	187	126	126	592	137	410	59			56	60		8.90					38.2
10	1.45	81	187	126	127	602	137	409	58	67	55	56	60		7.70					44.2
11	1.50	81	187	129	128	602	138	405	53			57	60		7.50					45.3
12	1.55	81	187	127	129	610	139	403	51			58	60		7.77					43.7
13	2.00	81	186	125	128	612	137	400	49	67	55	58	60	340#	5.90					57.7
Average		81	187	129	127	612	139	411	59	67	54	56	60	340#/Hr.	6.32	0				46.6

NON-CONDENSING.

STEAM - 50° Superheat.

LOAD - $\frac{1}{4}$.

H.P	Wg't. on Scales.	B.H.P	Mech. Eff.	Water-rate.	
				B.H.P	I.H.P
.3	145#	25.6	87.5	29.0	25.3
.0		25.0	83.4	29.7	24.7
.8		24.9	86.5	29.8	25.7
.9		25.0	81.0	29.6	24.0
.8		24.9	86.5	29.8	25.8
.2		25.1	80.5	29.6	23.8
.2		25.4	86.8	29.2	25.4
.1		25.6	88.0	29.0	25.5
.8		25.3	85.0	29.3	24.9
.5		25.3	89.0	29.3	25.8
.6		25.4	83.0	29.2	24.3
.4		25.4	89.5	29.2	26.1
.2		25.4	87.0	29.2	25.4
.8		25.6	89.0	29.0	25.8
.5		25.4	83.2	29.2	24.2
.9		25.4	87.9	29.2	25.7
.8		25.4	88.2	29.2	25.8
.1		25.4	79.1	29.2	23.1
.3		25.4	86.6	29.2	25.3
.5		25.3	85.7	29.3	25.2
.3		25.4	86.6	29.2	25.3
.9		25.4	88.0	29.2	25.7
.4		25.4	83.5	29.2	24.4
.1		25.3	84.0	29.3	24.6
.2		25.4	87.0	29.2	25.4
.6	145#	25.3	85.7	29.3	25.1

WATER RATE OF CORLISS ENGINE.—

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

PRESENTED BY { Weston B. Lazear. Franklin G. Rogers.

NO. OF TEST.- 5.
DATE.- May 1, 1907.

NON-CONDENSING.
STEAM- 50° Superheat.
LOAD- $\frac{1}{4}$.

No	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wgt. Steam Condensed	I.H.P.	Wg't. on Scales.	B.H.P.	Mech. Eff.	Water-rate.	
		Engine.	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng Room	Outside	Inflow	Outflow						B.H.P.	I.H.P.
1	2.10	80	185	127	128	626	136	411	60	67	57	59	68		29.3	145#	25.6	87.5	29.0	25.3
2	2.15	79	181	126	126	618	135	422	72			59	67		30.0		25.0	83.4	29.7	24.7
3	2.20	78	180	125	123	596	132	429	81			59	71		28.8		24.9	86.5	29.8	25.7
4	2.25	78	181	125	124	670	134	433	84	67	56	59	72		30.9		25.0	81.0	29.6	24.0
5	2.30	78	180	127	128	540	136	433	82			59	68		28.8		24.9	86.5	29.8	25.8
6	2.35	78	182	125	125	520	134	430	81			59	69		31.2		25.1	80.5	29.6	23.8
7	2.40	79	184	125	126	506	135	424	74			59	67	403#	29.2		25.4	86.8	29.2	25.4
8	2.45	80	185	126	128	520	137	418	67	62	56	59	68		29.1		25.6	88.0	29.0	25.5
9	2.50	80	183	125	125	530	133	411	62			59	68		29.8		25.3	85.0	29.3	24.9
10	2.55	80	183	124	123	510	133	405	56			59	70		28.5		25.3	89.0	29.3	25.8
11	3.00	80	184	126	126	500	134	402	53	63	56	58	67		30.6		25.4	83.0	29.2	24.3
12	3.05	80	184	124	125	510	134	399	50			58	68		28.4		25.4	89.5	29.2	26.1
13	3.10	80	184	131	130	504	140	396	43			58	66	345	29.2		25.4	87.0	29.2	25.4
14	3.15	80	185	130	128	514	136	394	43	68	56	58	67		28.8		25.6	89.0	29.0	25.8
15	3.20	80	184	130	129	524	139	394	42			58	66		30.5		25.4	83.2	29.2	24.2
16	3.25	80	184	130	130	560	139	393	43			58	66		28.9		25.4	87.9	29.2	25.7
17	3.30	80	184	126	127	558	136	394	43	66	55	58	66		28.8		25.4	88.2	29.2	25.8
18	3.35	80	184	128	128	598	135	398	48			58	67		32.1		25.4	79.1	29.2	23.1
19	3.40	80	184	122	123	596	131	400	52			58	67	376	29.3		25.4	86.6	29.2	25.3
20	3.45	80	183	123	123	590	133	406	57	66	55	60	69		29.5		25.3	85.7	29.3	25.2
21	3.50	80	184	123	123	588	132	409	61			60	69		29.3		25.4	86.6	29.2	25.3
22	3.55	80	184	121	120	584	130	412	65			60	70		28.9		25.4	88.0	29.2	25.7
23	4.00	80	184	123	122	578	132	414	66	66	56	60	71		30.4		25.4	83.5	29.2	24.4
24	4.05	79	183	127	128	540	136	416	65			59	67		30.1		25.3	84.0	29.3	24.6
25	4.10	80	184	127	128	528	136	416	65	67	53	58	66	360	29.2		25.4	87.0	29.2	25.4
Average		80	184	126	126	556	135	410	61	66	56	59	68	742#/Hr.	29.6	145#	25.3	85.7	29.3	25.1

NON-CONDENSING.

STEAM- 50° Superheat.

LOAD- $\frac{1}{2}$.

N.H.P.	Wg't. on Scales.	B.H.P.	Mech. Eff.	Water-rate.	
				I.H.P.	B.H.P.
18.5	250#	43.1	89.0	23.0	25.9
26.1		43.3	94.0	24.2	25.8
38.8		43.3	88.9	22.9	25.8
47.1		43.3	90.5	23.7	25.8
57.1		43.3	90.5	23.7	25.8
68.3		43.1	89.2	23.1	25.9
76.0		42.8	93.1	24.2	26.1
86.5		42.6	91.8	24.0	26.2
95.3		42.8	94.5	24.6	26.1
105.9		43.1	94.0	24.3	25.9
115.5		43.1	94.7	24.5	25.9
125.8		43.3	94.5	24.3	25.8
135.6		43.3	95.0	24.4	25.8
146.3		43.1	93.1	24.1	25.9
154.8		42.8	95.5	24.8	26.1
166.7		42.6	91.2	23.1	26.2
178.2		42.8	89.0	23.1	26.1
186.1		43.1	93.5	24.2	25.9
197.1		42.8	89.5	23.7	26.1
204.7		42.8	95.8	24.9	26.1
216.3		43.6	94.2	24.1	25.6
227.5		43.6	92.0	23.5	25.6
238.8		43.1	88.4	22.9	25.9
246.9		43.1	92.0	23.8	25.9
258.4		43.1	89.0	23.0	25.9
16.7	250	43.0	91.8	23.8	26.9

WATER RATE OF CORLISS ENGINE.—

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

PRESENTED BY { Weston B. Lazear. Franklin G. Rogers.

NO. OF TEST.- 6.
DATE.- May 4. 1907.

NON-CONDENSING.
STEAM- 50° Superheat.
LOAD- ½.

No.	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wgt. Steam Condensed	I.H.P.	Wgt. on Scales.	B.H.P.	Mech. Eff.	Water-rate.	
		Engine.	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng Room	Outside	Inflow	Outflow						I.H.P.	B.H.P.
1	3.30	79	181	125	121	470	124	408	64	62	57	56	69		48.5	250#	43.1	89.0	23.0	25.9
2	3.35	79	182	124	120	468	125	404	60			56	70		46.1		43.3	94.0	24.2	25.8
3	3.40	79	182	124	120	464	125	400	56			57	70		48.8		43.3	88.9	22.9	25.8
4	3.45	79	182	122	118	470	125	400	56	62	57	39	89		47.1		43.3	90.5	23.7	25.8
5	3.50	79	182	122	118	472	125	400	56			61	76		47.1		43.3	90.5	23.7	25.8
6	3.55	78	181	120	115	480	125	398	54			64	74		48.3		43.1	89.2	23.1	25.9
7	4.00	78	180	118	112	502	120	401	60	62	57	60	77		46.0		42.8	93.1	24.2	26.1
8	4.05	78	179	116	112	492	120	406	65			59	75	733#	46.5		42.6	91.8	24.0	26.2
9	4.10	78	180	120	115	500	125	408	64			58	75		45.3		42.8	94.5	24.6	26.1
10	4.15	78	181	120	115	498	125	410	66	61	54	57	71		45.9		43.1	94.0	24.3	25.9
11	4.20	78	181	118	114	484	120	410	69			57	71		45.5		43.1	94.7	24.5	25.9
12	4.25	79	182	120	116	474	125	402	58			57	71		45.8		43.3	94.5	24.3	25.8
13	4.30	79	182	124	120	470	125	402	58	61	54	57	71		45.6		43.3	95.0	24.4	25.8
14	4.35	78	181	121	118	470	125	400	56			57	71		46.3		43.1	93.1	24.1	25.9
15	4.40	78	180	120	116	474	120	398	57			57	71		44.8		42.8	95.5	24.8	26.1
16	4.45	78	179	116	111	472	120	398	57	62	54	57	71	652#	46.7		42.6	91.2	23.1	26.2
17	4.50	78	180	118	114	466	120	396	55			57	71		48.2		42.8	89.0	23.1	26.1
18	4.55	78	181	120	115	468	120	396	55			57	71		46.1		43.1	93.5	24.2	25.9
19	5.00	78	180	118	115	480	120	396	55	61	52	57	71		47.1		42.8	89.5	23.7	26.1
20	5.05	78	180	120	115	482	120	399	58			57	71		44.7		42.8	95.8	24.9	26.1
21	5.10	78	183	120	116	480	125	401	57			57	71	465#	46.3		43.6	94.2	24.1	25.6
22	5.15	80	183	120	116	478	125	402	58	68	52	57	71		47.5		43.6	92.0	23.5	25.6
23	5.20	80	181	120	116	476	125	402	58			57	71		48.8		43.1	88.4	22.9	25.9
24	5.25	78	181	120	116	474	125	402	58			57	71		46.9		43.1	92.0	23.8	25.9
25	5.30	79	181	120	116	472	125	400	56	70	52	57	71	376	48.4		43.1	89.0	23.0	25.9
Average		79	181	120	116	477	123	402	59	63	54	59	73	1113#/Hr.	46.7	250	43.0	91.8	23.8	26.9

NON-CONDENSING.

STEAM- 90° Superheat.

LOAD - 0

No.	I.H.P	Wg't. on Scales.	B.H.P.	Mech. Eff.	Water-rate	
					B.H.P.	I.H.P.
1	6.35	0#				49.5
2	[illegible]					[illegible]
3	5.15					61.0
4	5.33					59.0
5	6.98					45.0
6	6.12					51.2
7	6.63					47.4
8	5.12					61.3
9	5.86					53.6
10	6.29					50.0
11	6.98					45.7
12	6.05					51.9
13	5.60					56.0
Aver	6.04	0				52.5

WATER RATE
OF
CORLISS ENGINE.-

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

NO. OF TEST.- 7.
DATE.- May 4. 1907.

PRESENTED BY { Weston B. Lagear. Franklin G. Rogers.

NON-CONDENSING.
STEAM- 90° Superheat.
LOAD - 0

No.	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wg't. Steam Condensed	I.H.P	Wg't. on Scales	B.H.P.	Mech. Eff.	Water-rate	
		Engine	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng Room	Outside	Inflow	Outflow						B.H.P.	I.H.P.
1	1.55	87	200	135	133	520	140	431	78	63	56	55	60		6.35	0#				49.5
2	2.00	90	206	135	133	524	140	424	71			56	60							
3	2.05	79	184	129	126	540	135	422	72			56	60		5.15					61.0
4	2.10	78	181	125	123	550	130	423	76			56	60		5.33					59.0
5	2.15	86	197	130	127	558	130	428	81	63	55	56	60		6.98					45.0
6	2.20	80	184	130	128	560	130	429	82			56	60		6.12					51.2
7	2.25	80	182	128	125	560	130	429	82			56	60		6.63					47.4
8	2.30	80	187	130	126	560	130	429	82	62	54	56	60		5.12					51.3
9	2.35	81	189	127	125	560	130	429	82			56	60		5.86					53.6
10	2.40	82	189	125	123	558	130	429	82			56	60		6.29					50.0
11	2.45	82	189	129	126	560	130	429	82	62	53	56	60		698					45.7
12	2.50	81	189	134	132	560	140	430	77			56	60		6.05					51.9
13	2.55	81	189	136	134	558	141	432	79	62	55	56	60	314#	5.60					56.0
Average		82	190	130	127	551	134	428	79	62	55	56	60	314#/Hr.	6.04	0				52.5

NON-CONDENSING.
STEAM- 90°- Superheat
LOAD- $\frac{1}{4}$

No.	I.H.P	Wgt. on Scales.	B.H.P	Mech. Eff.	Water-rate.	
					B.H.P	I.H.P
1.	29.4	145#	26.8	91.1	25.5	23.2
2.	29.1		26.8	92.1	25.5	23.4
3.	28.7		26.8	93.1	25.5	23.8
4.	28.5		26.8	94.0	25.5	24.0
5.	31.1		26.8	86.0	25.5	22.0
6.	30.9		26.6	86.0	25.7	22.1
7.	27.9		26.4	94.6	25.9	24.4
8.	28.4		25.8	91.0	26.4	24.0
9.	27.4		25.6	94.0	26.7	24.9
10.	29.8		25.7	86.3	26.6	22.9
11.	30.7		25.7	83.7	26.6	22.2
12.	29.2		25.7	88.0	26.6	23.4
13.	29.5		25.7	87.2	26.6	23.1
14.	27.2		25.6	94.0	26.7	25.1
15.	29.8		25.6	86.0	26.7	22.9
16.	27.8		25.6	92.0	26.7	24.6
17.	26.9		25.4	94.4	26.9	25.4
18.	28.1		25.4	90.5	26.9	24.3
19.	28.3		25.4	89.8	26.9	24.1
20.	27.4		25.7	94.0	26.6	24.9
21.	29.0		25.8	89.0	26.5	23.5
22	28.7		25.8	90.0	26.5	23.8
23	28.1		25.8	92.0	26.5	24.3
24	27.2		25.8	94.8	26.5	25.1
25	29.8		25.7	86.3	26.6	22.9
Ave	28.7	145#	26.0	90.4	26.3	23.8

WATER RATE
OF
CORLISS ENGINE.—

THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

PRESENTED BY { Weston B. Lagcar. Franklin G. Rogers.

NO. OF TEST.- 8.
DATE.- May 4. 1907

NON-CONDENSING.
STEAM- 90°- Superheat
LOAD- $\frac{1}{4}$

No	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superh't.	Temperatures.		Cooling Water.		Wgt. Steam Condensed	I.H.P	Wgt. on Scales.	B.H.P	Mech. Eff.	Water-rate.	
		Engine.	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng Room	Outside	Inflow	Outflow						B.H.P	I.H.P
1.	11.50	84	194	130	126	546	137	442	91	72		54	63		29.4	145#	26.8	91.1	25.5	23.2
2.	11.55	84	194	128	125	532	136	442	91			54	63		29.1		26.8	92.1	25.5	23.4
3.	12.00	84	194	130	127	528	135	438	88	72		54	63		28.7		26.8	93.1	25.5	23.8
4	12.05	84	194	134	130	526	135	436	86			54	63		28.5		26.8	94.0	25.5	24.0
5	12.10	83	194	124	119	538	128	434	88			54	63		31.1		26.8	86.0	25.5	22.0
6.	12.15	82	193	125	123	540	130	434	87	72		54	63		30.9		26.6	86.0	25.7	22.1
7	12.20	82	191	127	124	530	130	434	87			54	63	362#	27.9		26.4	94.6	25.9	24.4
8.	12.25	81	187	126	123	538	130	434	87			54	63		28.4		25.8	91.0	26.4	24.0
9.	12.30	80	185	125	120	544	128	433	87	72		54	63		27.4		25.6	94.0	26.7	24.9
10	12.35	80	186	124	121	544	128	434	88			54	63		29.8		25.7	86.3	26.6	22.9
11.	12.40	81	186	134	130	542	135	438	88			54	63		30.7		25.7	83.7	26.6	22.2
12	12.45	81	186	130	127	536	135	438	88	70		55	63		29.2		25.7	88.0	26.6	23.4
13.	12.50	81	186	130	126	542	135	434	84			55	63		29.5		25.7	87.2	26.6	23.1
14	12.55	80	185	129	124	548	130	434	87			55	64		27.2		25.6	94.0	26.7	25.1
15.	1.00	80	185	130	126	548	133	435	86	65		55	64		29.8		25.6	86.0	26.7	22.9
16.	1.05	80	185	130	126	544	135	436	86			55	63		27.8		25.6	92.0	26.7	24.6
17	1.10	80	184	129	125	552	132	437	89			55	63		26.9		25.4	94.4	26.9	25.4
18	1.15	80	184	129	125	560	132	438	90	64		55	64	620#	28.1		25.4	90.5	26.9	24.3
19	1.20	80	184	130	126	556	133	442	93			55	64		28.3		25.4	89.8	26.9	24.1
20	1.25	80	186	132	129	552	135	443	92			55	64		27.4		25.7	94.0	26.6	24.9
21.	1.30	80	187	134	130	552	137	443	92	64		55	64		29.0		25.8	89.0	26.5	23.5
22	1.35	80	187	134	131	540	138	442	90			55	64		28.7		25.8	90.0	26.5	23.8
23	1.40	81	187	134	130	540	137	439	89			56	64		28.1		25.8	92.0	26.5	24.3
24	1.45	81	187	135	131	550	138	440	87	63		56	64		27.2		25.8	94.8	26.5	25.1
25	1.50	81	186	135	130	540	137	439	88			56	64	383#	29.8		25.7	86.3	26.6	22.9
Average.		81	188	130	126	543	134	438	88	69		55	63	683#/Hr	28.7	145#	26.0	90.4	26.3	23.8

NON-CONDENSING.
STEAM- 90° Superheat.
LOAD- $\frac{1}{2}$

No.	I.H.P.	Wgt. on Scales.	B.H.P.	Mech. Eff.	Water-rate.	
					B.H.P.	I.H.P.
1.	52.4	285#	48.0	91.6	24.1	22.1
2.	50.3		47.7	94.8	24.2	23.0
3.	—		47.5	—	24.3	—
4.	52.8		48.0	91.0	24.1	21.9
5.	52.0		49.1	94.5	23.5	22.2
6.	53.8		50.1	93.2	23.1	21.5
7.	53.6		50.1	93.5	23.1	21.6
8.	53.1		49.9	94.0	23.2	21.8
9.	54.1		50.1	92.7	23.1	21.4
10.	53.1		50.1	94.4	23.1	21.8
11.	53.3		49.9	93.5	23.2	21.7
12.	52.8		49.9	94.5	23.2	21.9
13.	52.2		50.1	94.0	23.1	21.7
14.	53.8		50.1	93.3	23.1	21.5
15.	53.1		50.1	94.4	23.1	21.8
16.	54.5		50.4	92.5	22.9	21.2
17.	53.0		50.4	95.1	22.9	21.8
18.	53.0		50.4	95.1	22.9	21.8
19.	54.0		50.4	93.4	22.9	21.4
20.	54.3		50.7	93.4	22.8	21.3
21.	53.4		51.0	95.5	22.7	21.6
22.	54.2		51.2	94.5	22.6	21.3
23.	53.6		51.0	95.2	22.7	21.6
24.	56.0		51.0	91.0	22.7	20.7
25.	55.7		50.7	91.0	22.8	20.8
Av.	53.4	285#	50.0	95.6	23.2	21.6

WATER RATE
OF
CORLISS ENGINE.—

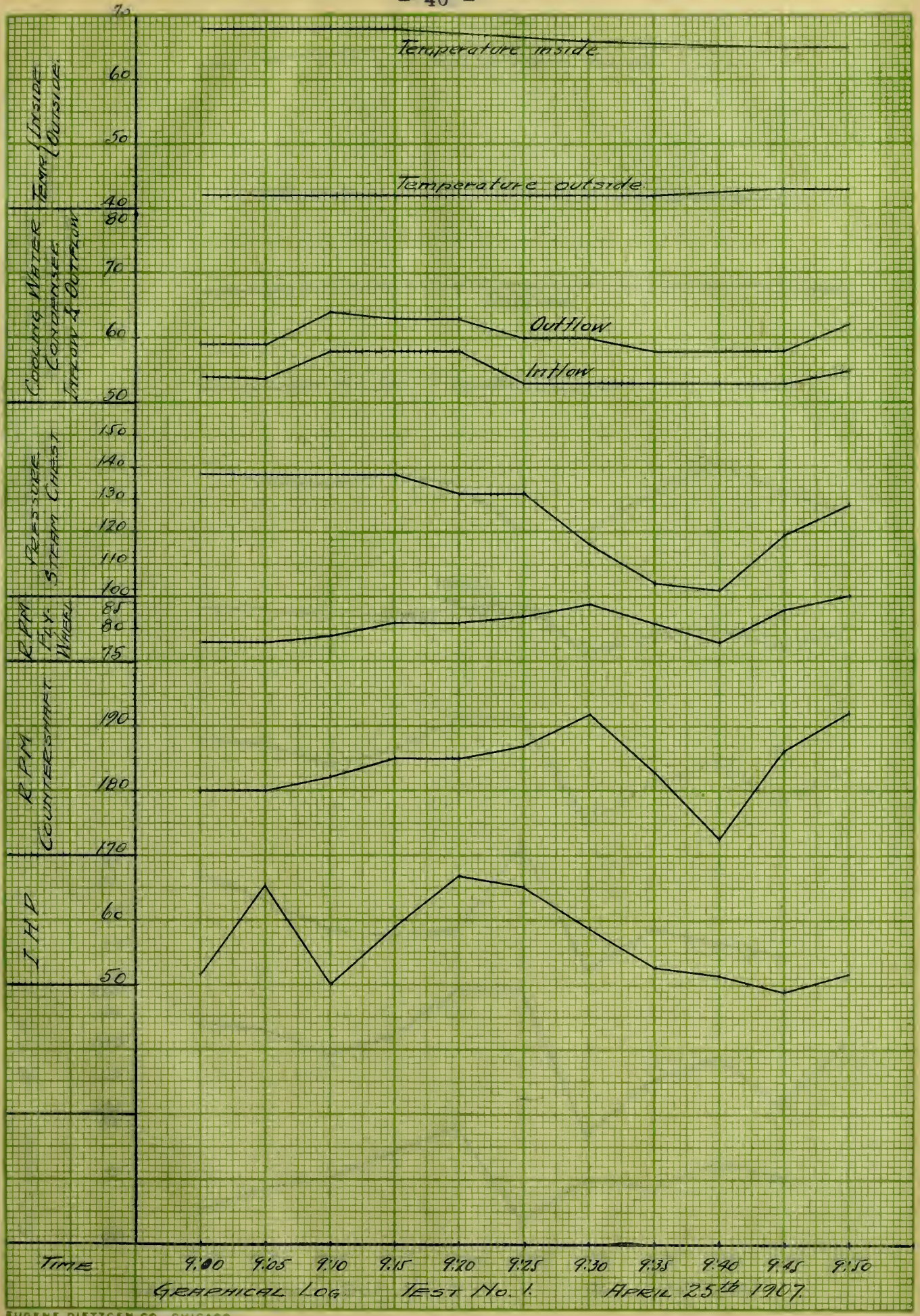
THESIS FOR DEGREE OF B.S. IN MECHANICAL ENGINEERING.

PRESENTED BY { Weston B. Lazear
Franklin G. Rogers.

NO. OF TEST.- 9.
DATE.- MAY 4 1907

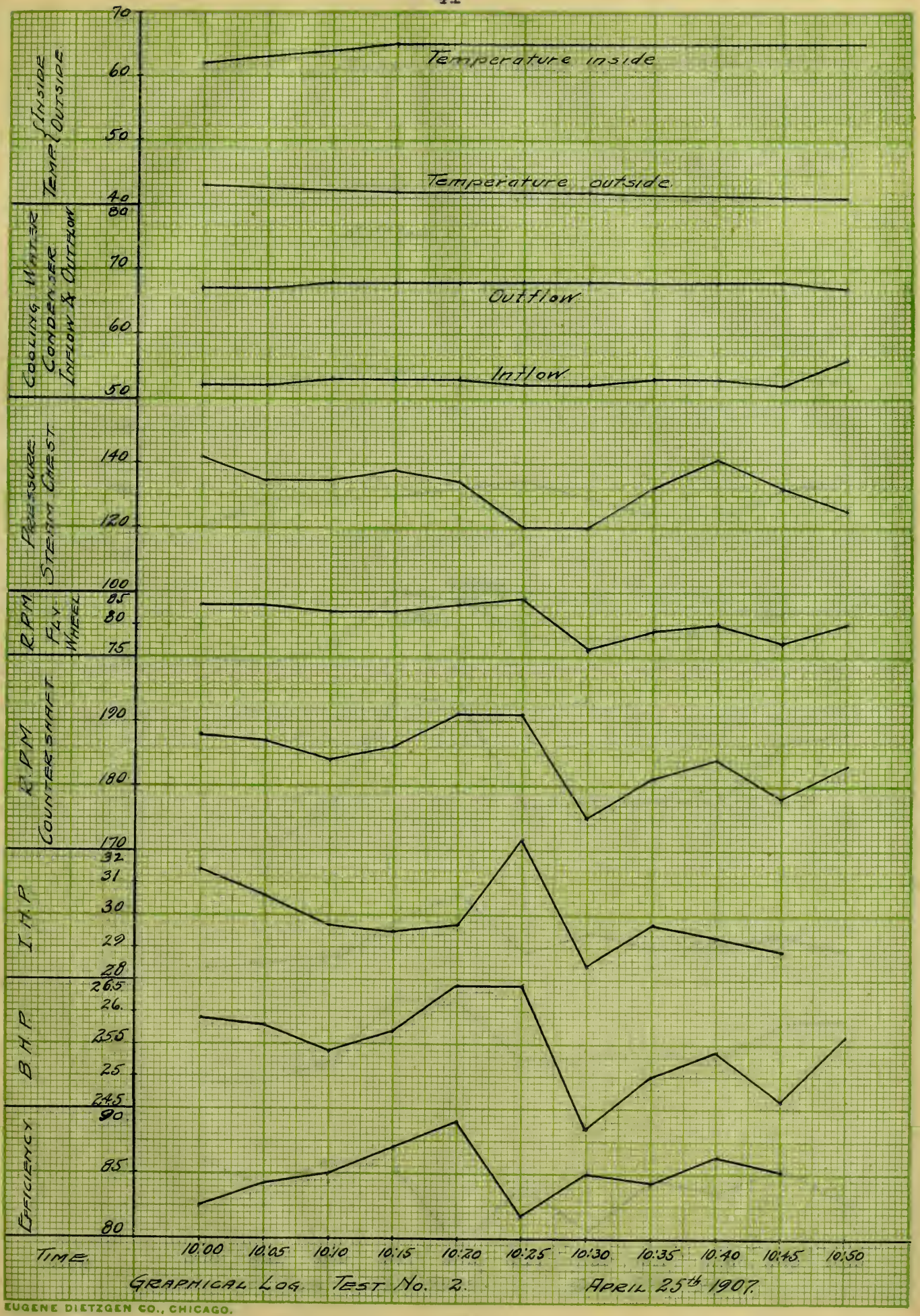
NON-CONDENSING.
STEAM- 90° Superheat
LOAD- 1/2

No	Time.	R.P.M.		Superheater.			Steam Chest.		Degrees of Superht.	Temperatures.		Cooling Water.		Wgt. Steam Condensed	I.H.P.	Wgt. on Scales.	B.H.P.	Mech. Eff.	Water-rate.	
		Engine.	Brake.	Press. Sat.	Press. Sup.	Temp. Sup.	Pressure	Temper.-		Eng Room	Outside	Inflow	Outflow						B.H.P.	I.H.P.
1.	9.45	77	177	132	129	510	135	426	76	60	45	50	63		52.4	285#	48.0	91.6	24.1	22.1
2.	9.50	76	176	131	127	512	135	426	76			50	63		50.3		47.7	94.8	24.2	23.0
3	9.55	76	175	126	121	570	125	424	80			51	64		—		47.5	—	24.3	—
4	10.00	77	177	120	126	540	130	424	97	59	50	51	64		52.8		48.0	91.0	24.1	21.9
5	10.05	79	181	126	123	540	130	436	89			51	64		52.0		49.1	94.5	23.5	22.2
6	10.10	80	185	127	124	552	130	438	91			52	65	502#	53.8		50.1	93.2	23.1	21.5
7.	10.15	80	185	126	122	550	130	442	95	63	50	52	65		53.6		50.1	93.5	23.1	21.6
8.	10.20	80	184	126	123	538	130	446	99			52	65		53.1		49.9	94.0	23.2	21.8
9.	10.25	81	185	127	124	532	130	444	97			52	65		54.1		50.1	92.7	23.1	21.4
10.	10.30	80	185	126	120	534	125	442	98	68	53	53	66		53.1		50.1	94.4	23.1	21.8
11.	10.35	80	184	120	116	530	125	440	96			53	66		53.3		49.9	93.5	23.2	21.7
12.	10.40	80	184	119	114	530	120	440	99			53	67	573#	52.8		49.9	94.5	23.2	21.9
13.	10.45	81	185	120	114	520	120	438	97	68	50	53	67		52.2		50.1	94.0	23.1	21.7
14.	10.50	81	185	119	113	518	120	438	97			53	67		53.8		50.1	93.3	23.1	21.5
15.	10.55	82	185	118	112	516	125	432	91			53	67		53.1		50.1	94.4	23.1	21.8
16.	11.00	80	186	120	115	525	120	434	93	68	52	53	68		54.5		50.4	92.5	22.9	21.2
17.	11.05	80	186	120	114	528	125	438	94			53	68		53.0		50.4	95.1	22.9	21.8
18	11.10	81	186	118	112	527	120	440	99			54	65		53.0		50.4	95.1	22.9	21.8
19.	11.15	81	186	120	116	540	120	440	99	70	52	54	68	660#	54.0		50.4	93.4	22.9	21.4
20.	11.20	81	187	132	127	530	125	441	97			53	67		54.3		50.7	93.4	22.8	21.3
21.	11.25	82	188	135	127	520	135	444	94			54	67		53.4		51.0	95.5	22.7	21.6
22.	11.30	81	189	135	130	532	135	442	92	70	54	54	67		54.2		51.2	94.5	22.6	21.3
23.	11.35	82	188	130	125	543	135	440	90			57	71		53.6		51.0	95.2	22.7	21.6
24.	11.40	82	188	126	122	548	130	442	95			55	69		56.0		51.0	91.0	22.7	20.7
25.	11.45	82	187	125	121	550	130	444	97	72	54	54	68	576#	55.7		50.7	91.0	22.8	20.8
Average.		81	184	125	121	531	127	438	92	66	51	53	66	1156#/Hr.	53.4	285#	50.0	95.6	23.2	21.6



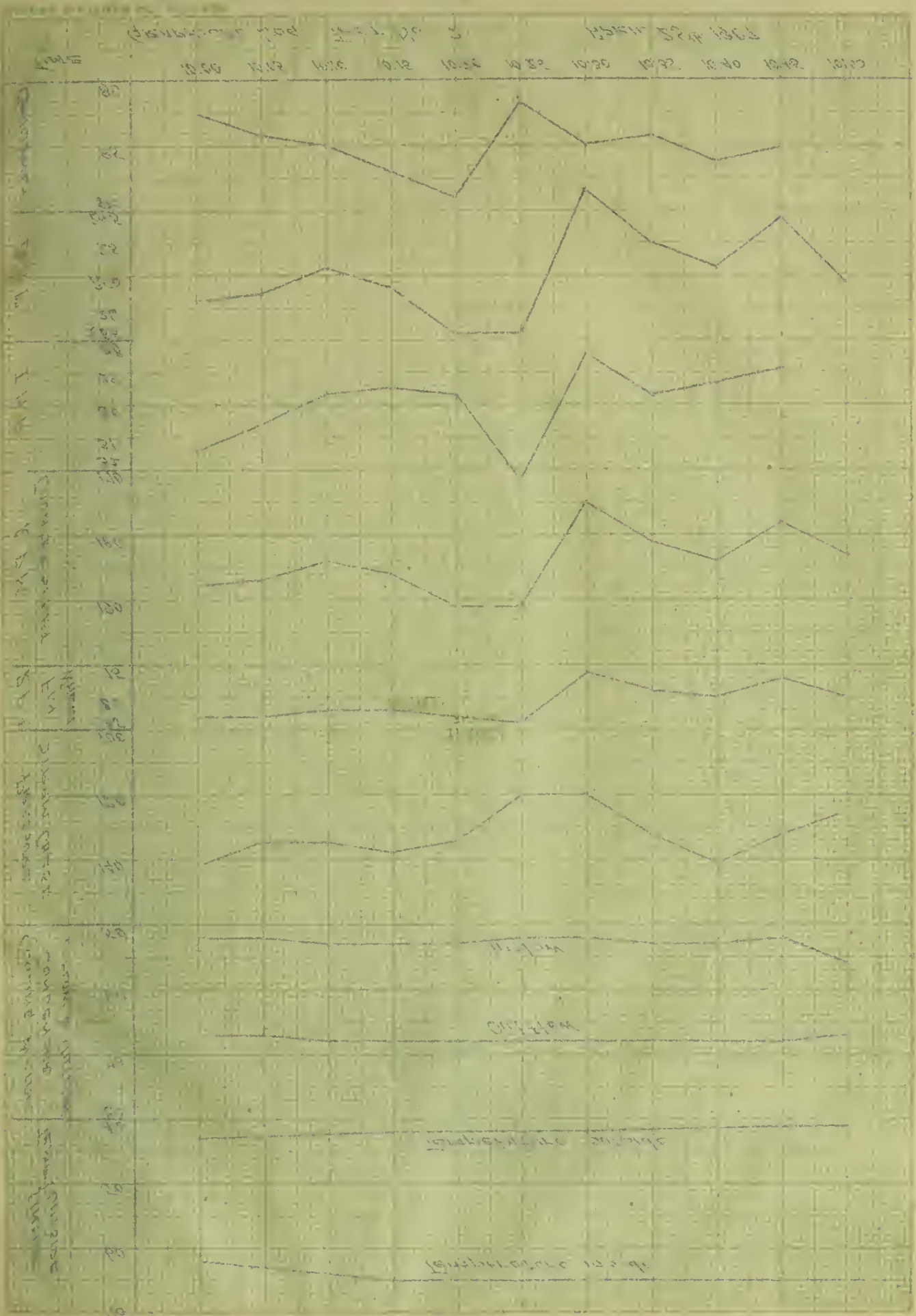
Graphical Log. Test No. 1. April 25th 1907.

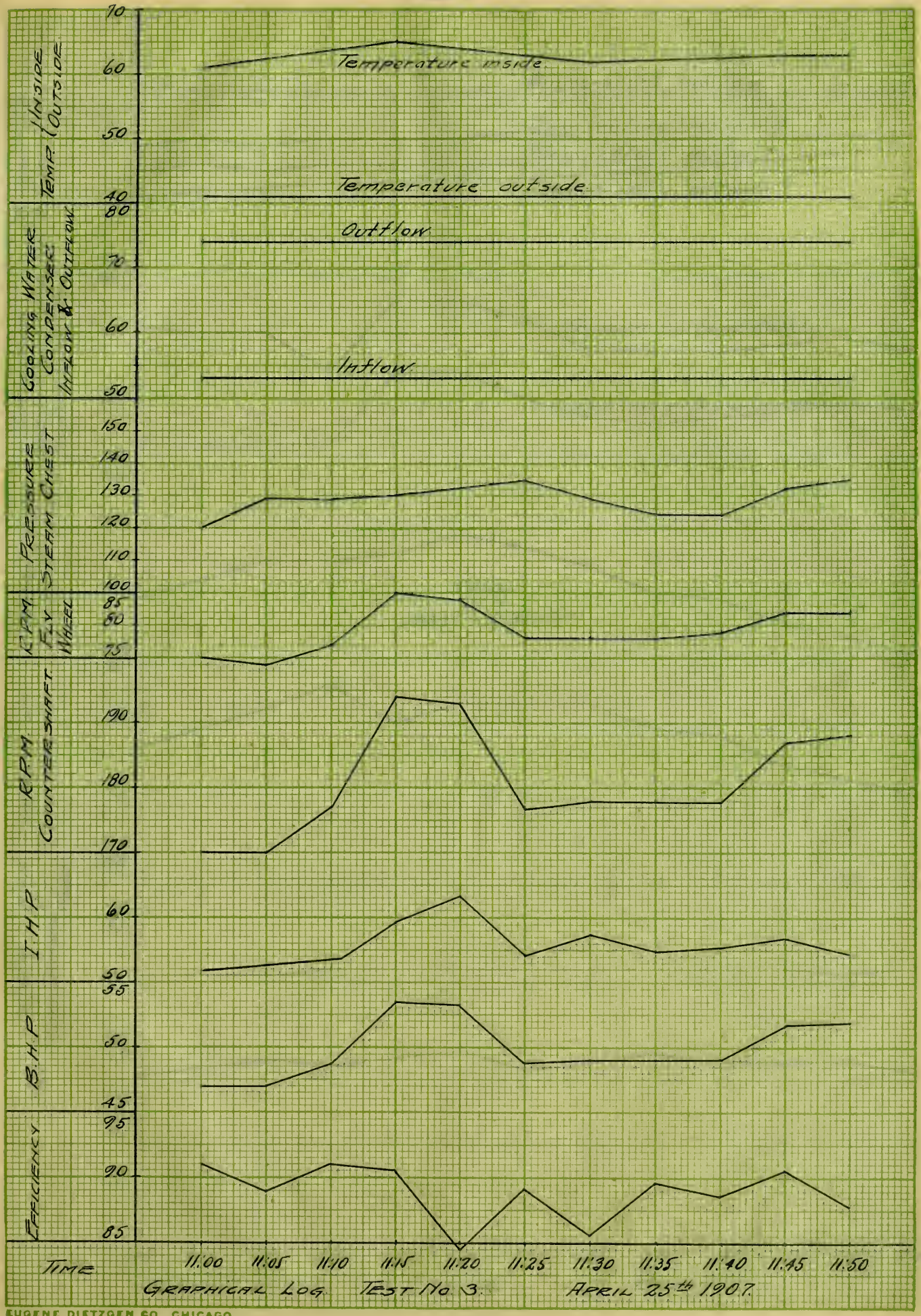
EUGENE DIETZGEN CO., CHICAGO.



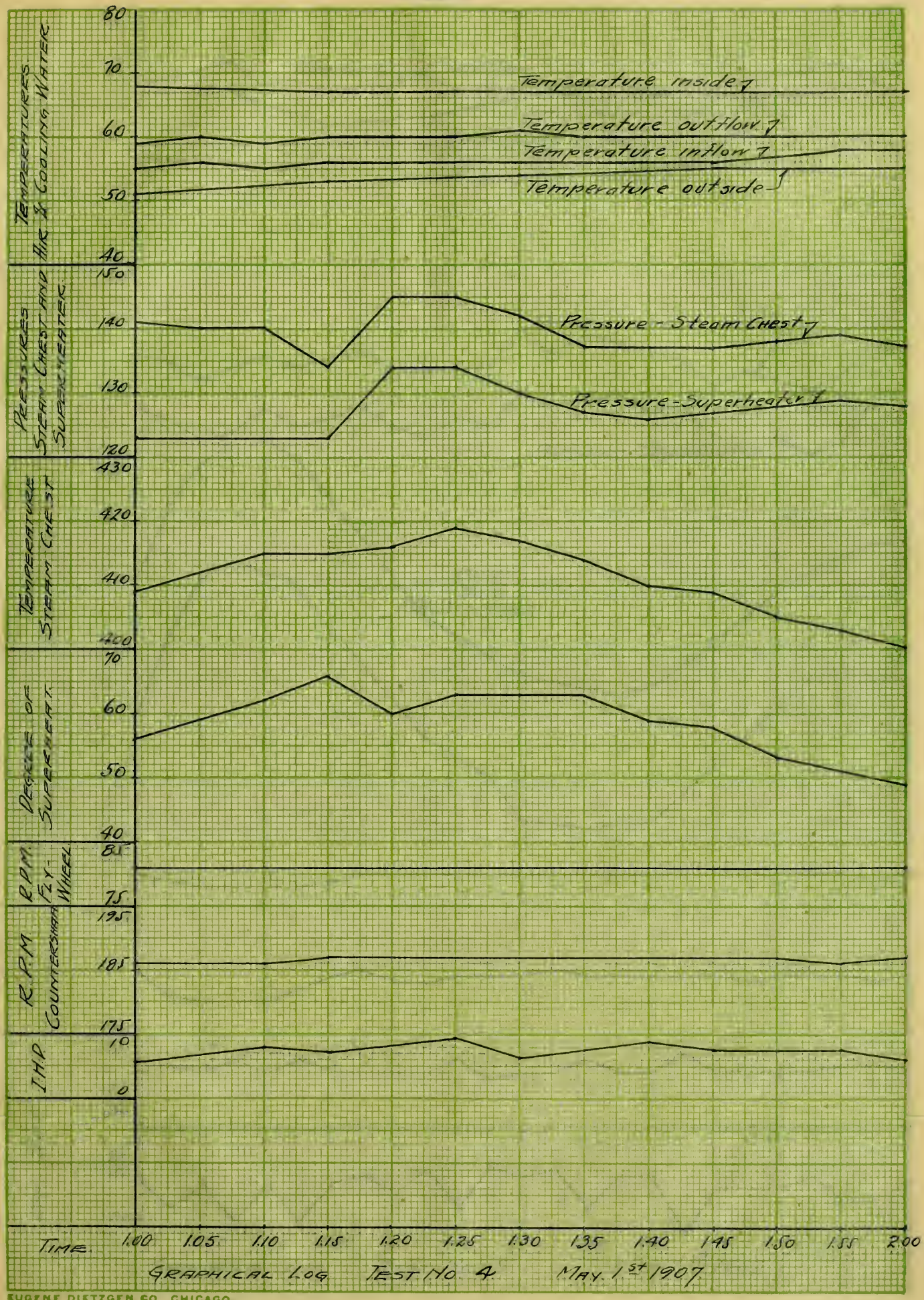
Graphical Log. Test No. 2. April 25th 1907.

EUGENE DIETZGEN CO., CHICAGO.



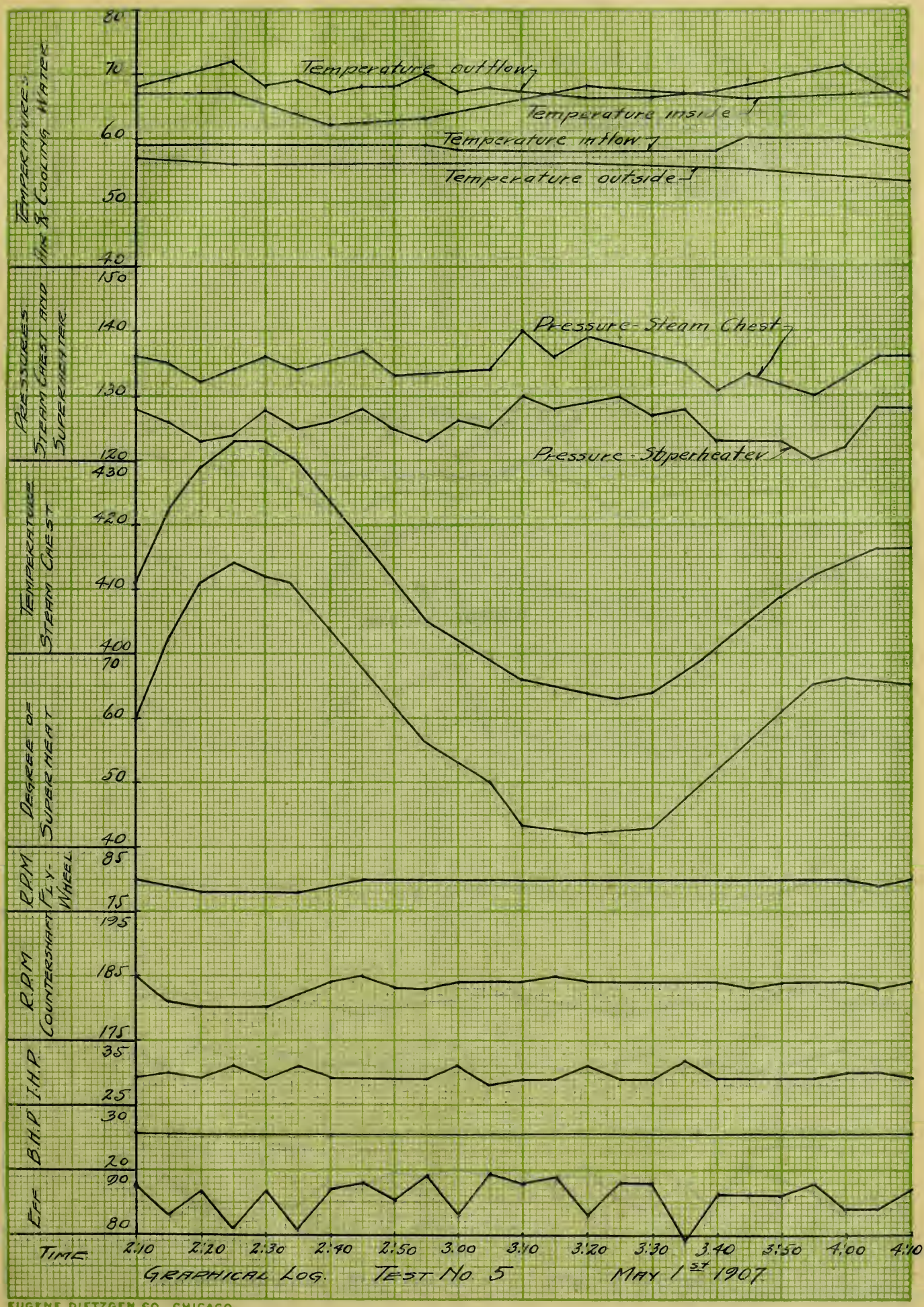


EUGENE DIETZGEN CO., CHICAGO.



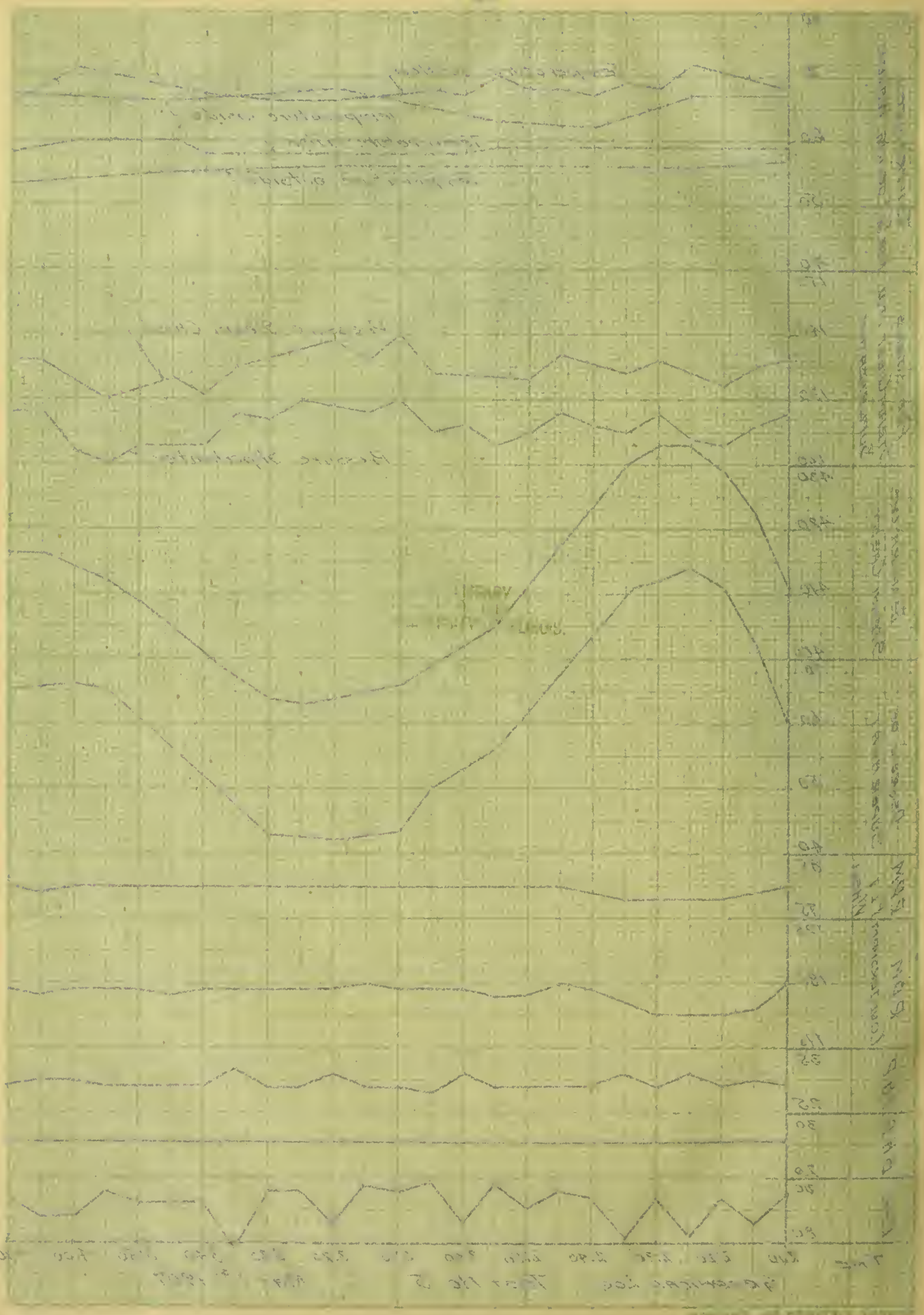
Graphical Log Test No. 4 May. 1st 1907

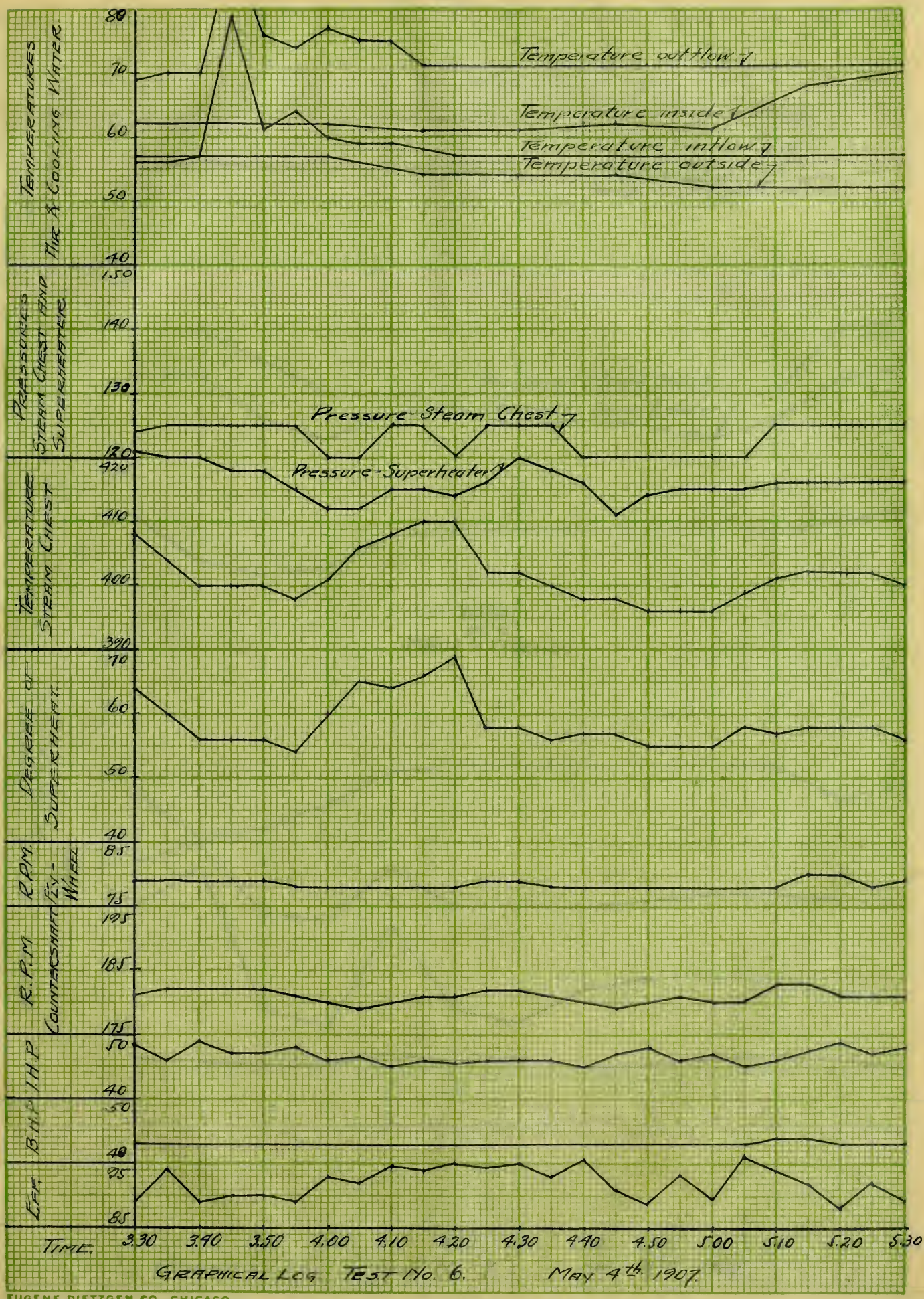
EUGENE DIETZGEN CO., CHICAGO.



GRAPHICAL LOG. TEST No. 5 MAY 1st 1907

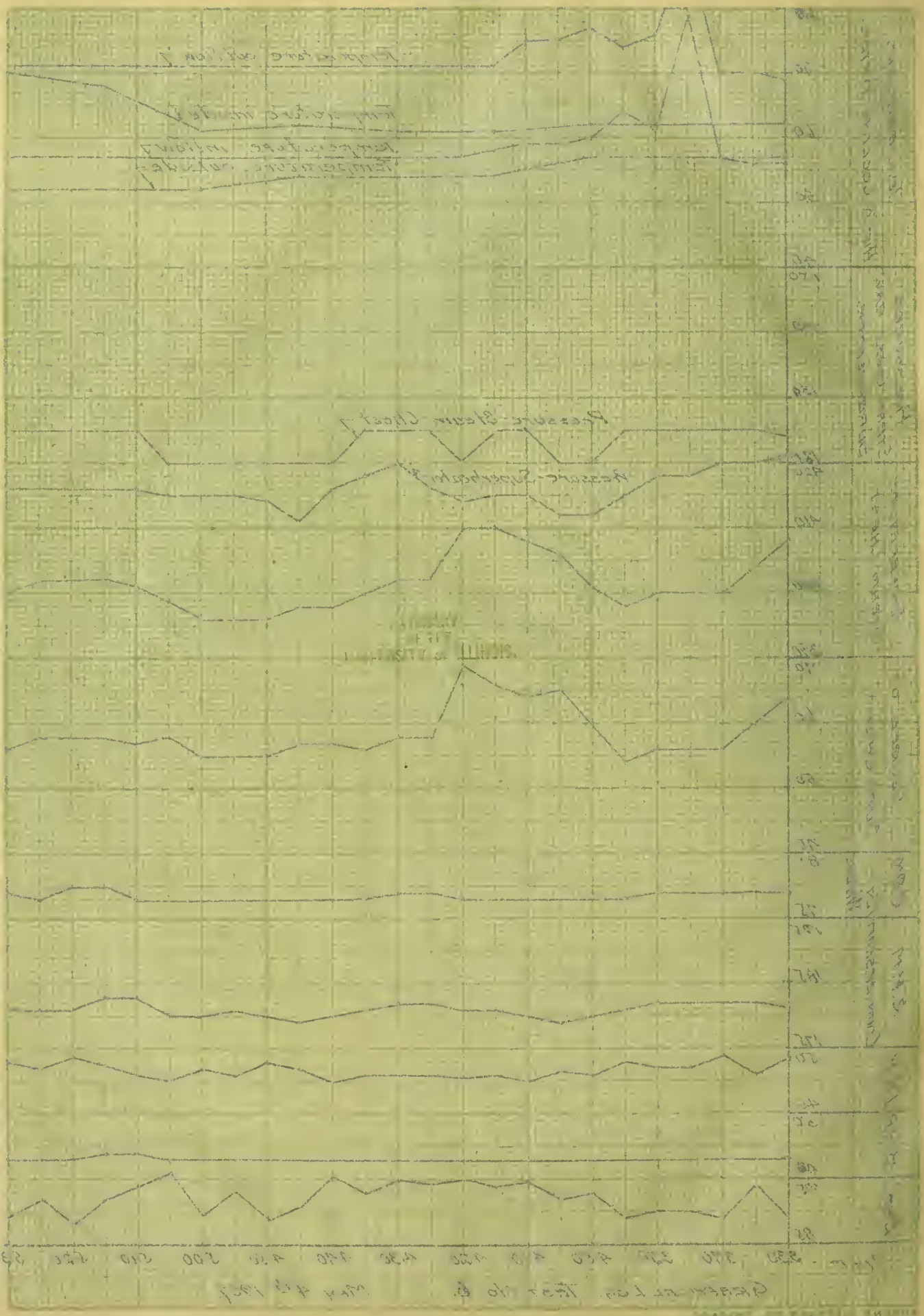
EUGENE DIETZGEN CO., CHICAGO.

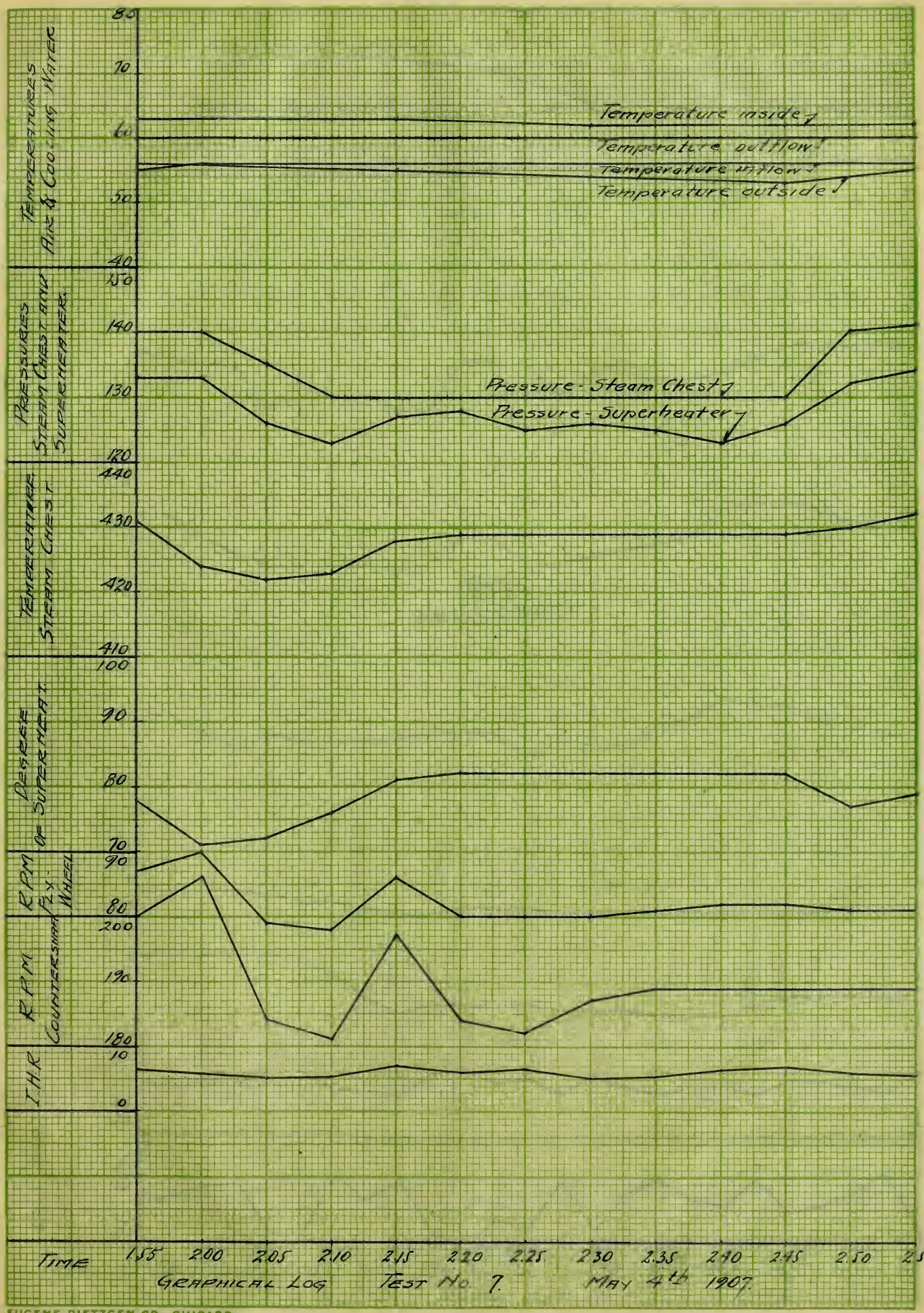




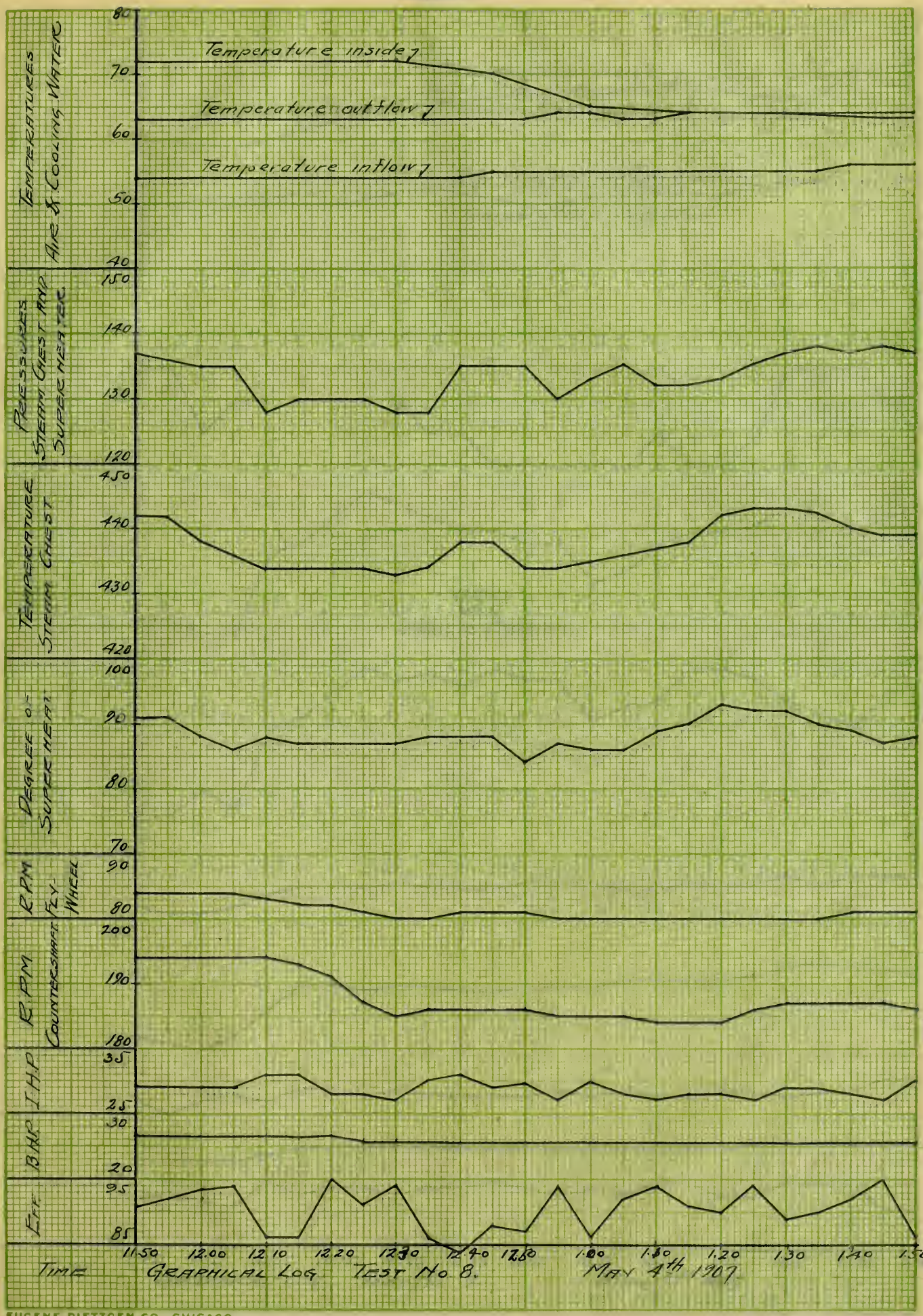
Graphical Log Test No. 6. May 4th. 1907.

EUGENE DIETZGEN CO., CHICAGO.

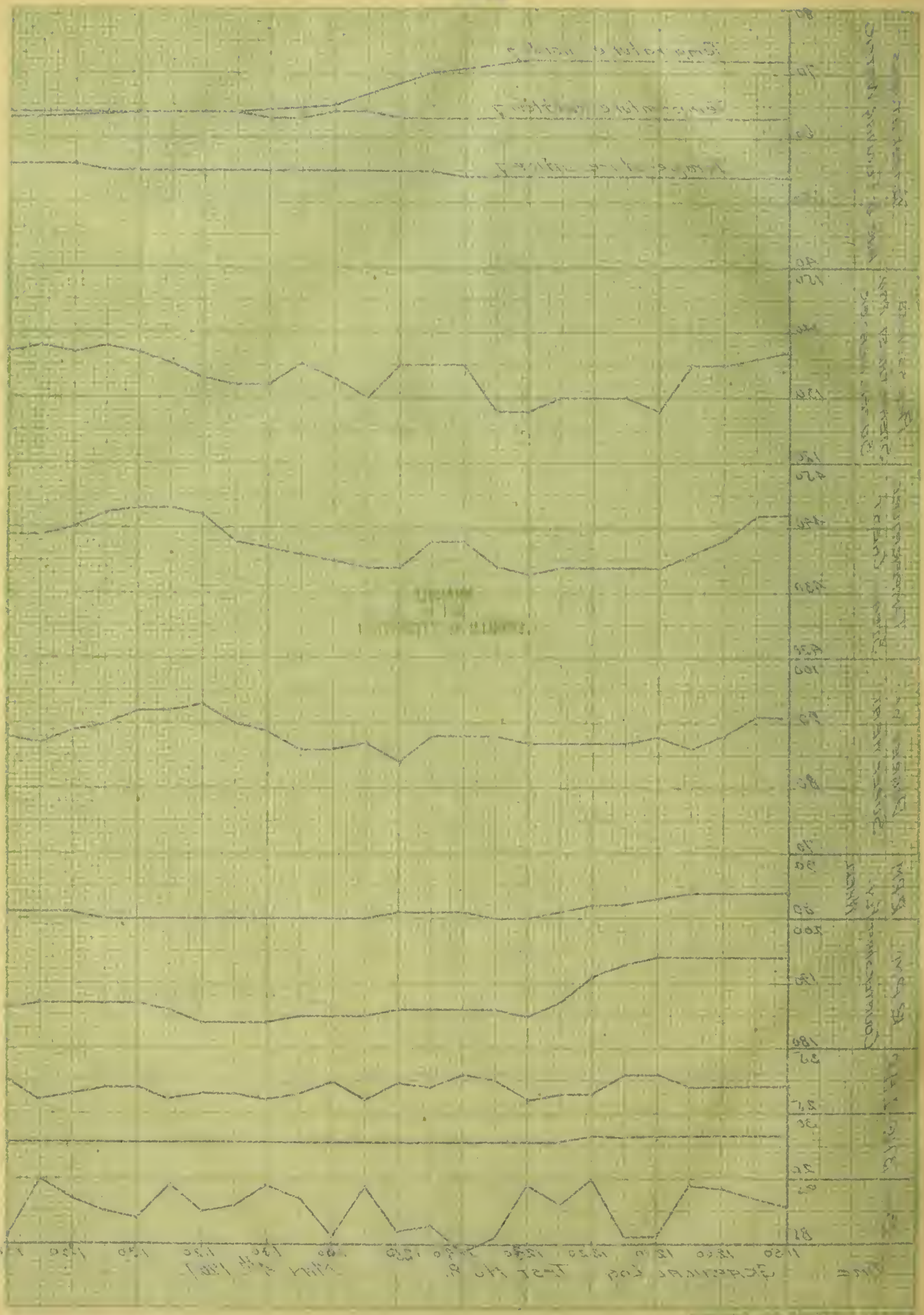


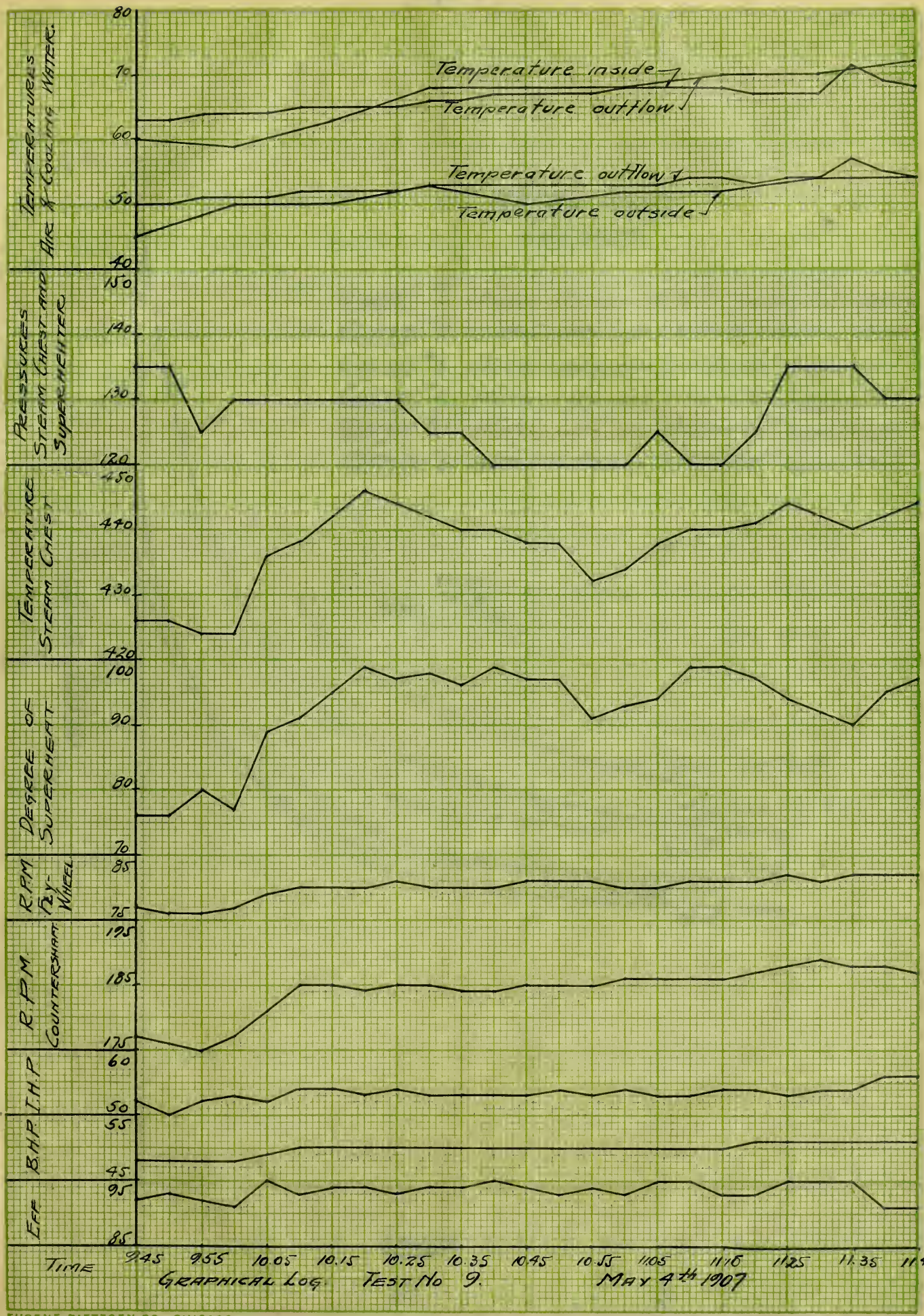


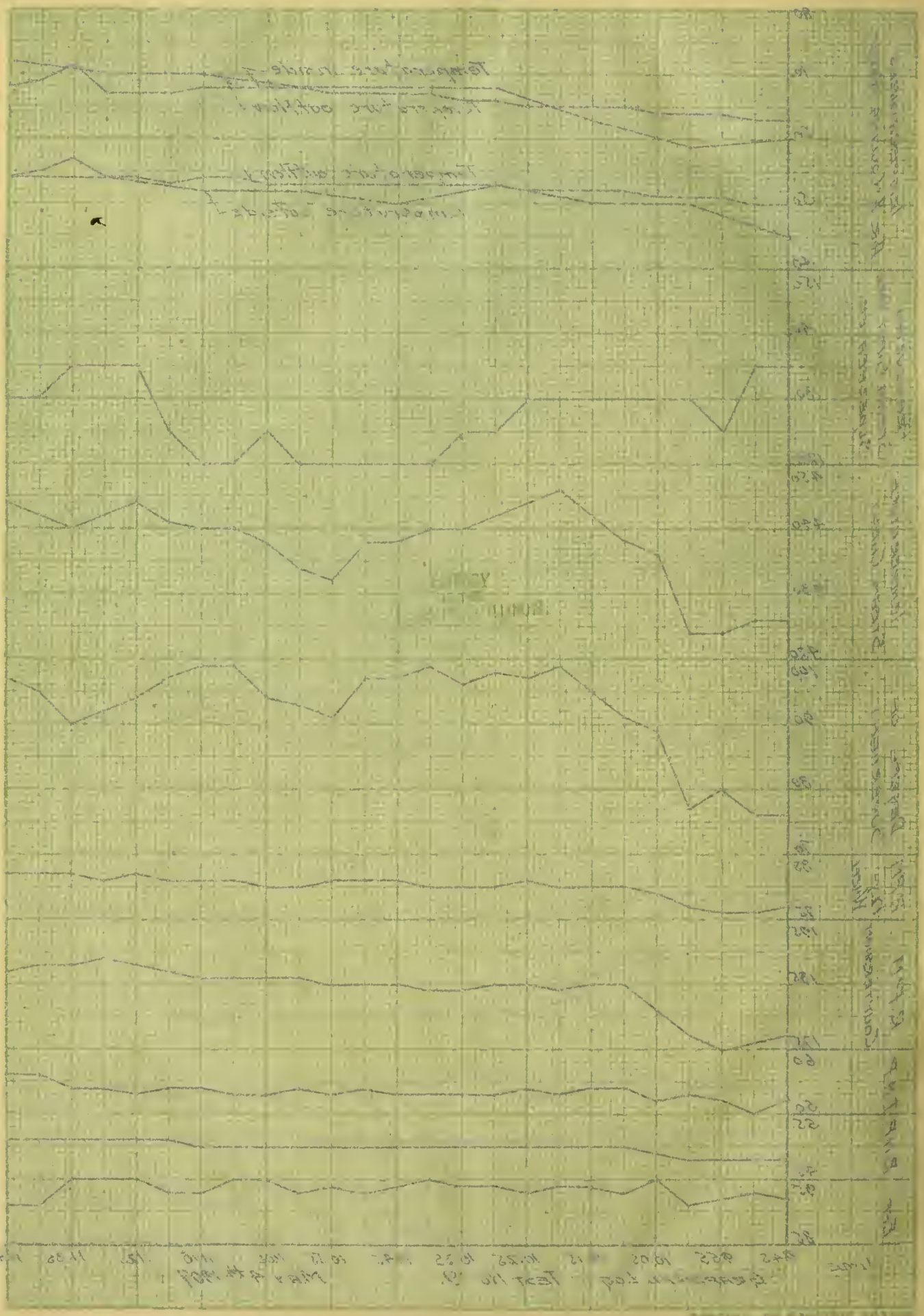
EUGENE DIETZGEN CO., CHICAGO.

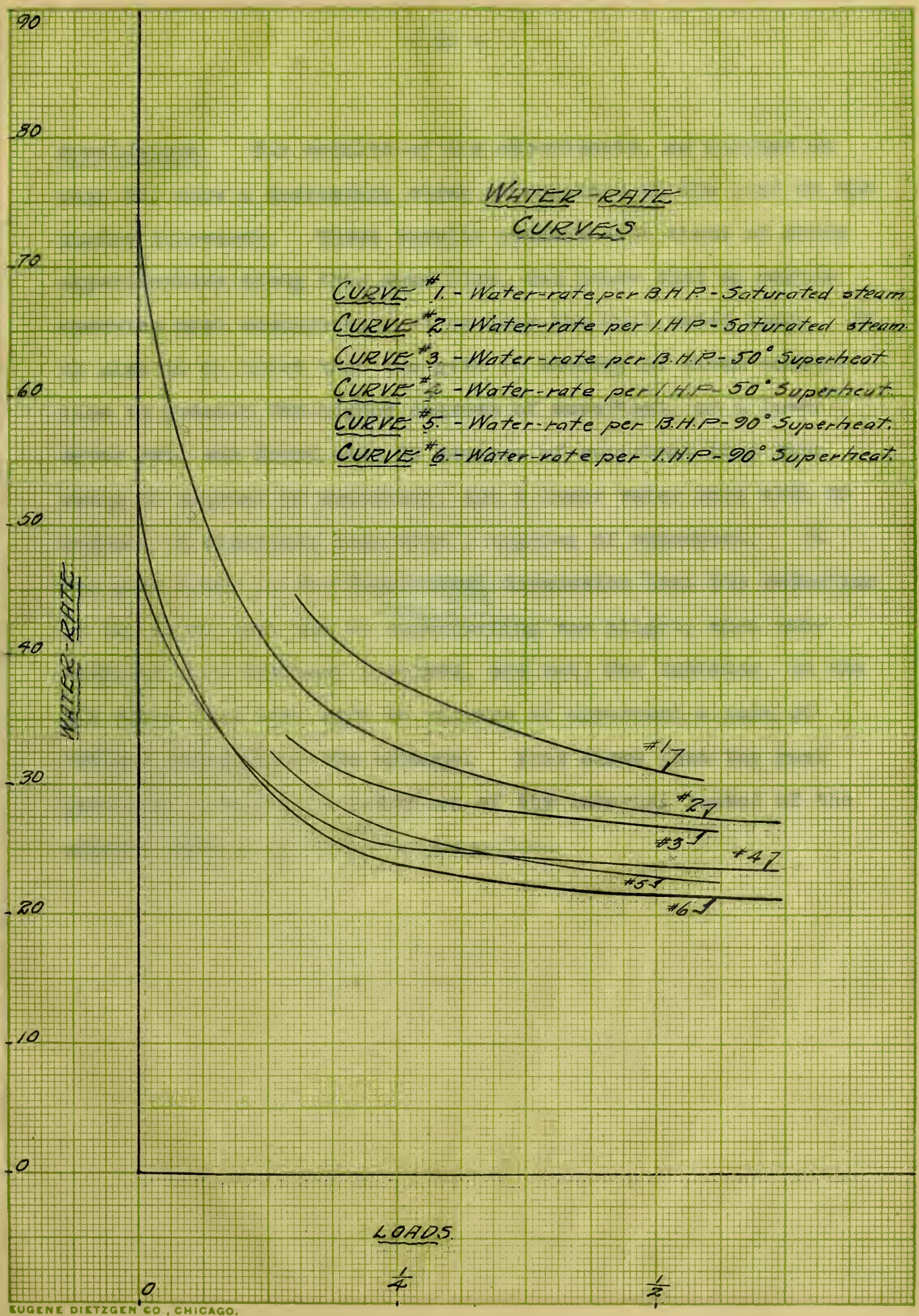


EUGENE DIETZGEN CO., CHICAGO.




Temperatures Air & Cooling Water.
80
70
60
50
40
Temperature inside
Temperature outflow
Temperature outflow
Temperature outside
Pressures Steam Chest and Superheater.
150
140
130
120
Temperature Steam Chest
450
440
430
420
Degree of Superheat.
100
90
80
70
R.P.M. Fly Wheel
85
75
R.P.M. Countershaft
195
185
175
I.H.P.
60
50
B.H.P.
55
45
Eff.
95
85
Time
9.45
9.55
10.05
10.15
10.25
10.35
10.45
10.55
11.05
11.15
11.25
11.35
11.45
Graphical Log. Test No 9. May 4th 1907
Eugene Dietzgen Co., Chicago.





EUGENE DIETZGEN CO., CHICAGO.

Conclusions. The results of the experiments, as plotted on page 49, show a decreasing steam consumption as the load on the engine increases. These results conform with those of other experimenters along this same line, and prove that an engine operates most economically at or near the full rated load. It is also to be noted that the gain in economy of steam consumption is greater for higher degrees of superheat, as a lower water rate was found with 90 degrees of superheat than when using 50 degrees of superheat, and a lower water rate with 50 degrees of superheat than with 0 degrees of superheat. It was also found in the theoretical discussion that the reduction in the water rate due to superheating was slight, when considering only the heat involved, and yet, for instance, in the 1/2 rated load test with 90 degrees of superheat a gain of 25% was obtained in the economy. This shows that the real gain must come from a reduction of the internal losses of the engine, such as cylinder condensation.

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